THE UNIVERSITY OF MICHIGAN
COLLEGE OF ENGINEERING
DEPARTMENT OF NAVAL ARCHITECTURE
AND MARINE ENGINEERING

2600 DRAPER RD., NORTH CAMPUS
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Report on ONR Workshop on Nonlinear Sea Loads and Ship Response: A Basis for Ship Structural Design

Location: Department of Naval Architecture and Marine Engineering University of Michigan Ann Arbor, Michigan 48109-2145

Date: July 7 - 8, 1994

The ONR Sea Loads-Ship Response (SLSR) program includes research areas related to nonlinear hydrodynamics, nonlinear dynamics, structural fatigue, elastic and plastic structural deformation, and a probabilistic or reliability-based analysis of ship structural design. Due to the diverse and multi-disciplinary nature of the project, program researchers were brought together at the University of Michigan to discuss the direction of their current and future research; the goal being to achieve a high level of coordination between the various efforts. This report contains copies of the presentations made at the workshop.

Thirty-one participants from various academic institutions, government laboratories and offices, and commercial companies attended. Presentations representing the state-of-the-art were made in the areas of hydrodynamic loading, structural analysis, design reliability, and simulation-based design.

Experts in hydrodynamics (SAIC, MIT, AMI, and UofM) explained that by using various nonlinear or partially nonlinear models, computer codes are capable of determining hydrodynamic loads, excluding bottom impact or flare slamming, in random seas. However, given the success of recent planing hull studies, the extension of planing hull hydrodynamics to the impact problem should be straightforward thus allowing for the complete hydrodynamic loads time history in extreme seas to be made. From these time histories, the design hydrodynamic and inertial loading events can be determined.

Structural experts (CDNSWC, NAVESEA, Ross and McNatt, and ABS) explained how the hydrodynamic and inertial loads are currently estimated and used in the structural design of ships, both naval and commercial. Due to the complexity of a ship's structure and the need for timely engineering answers, hydrodynamic load modeling is generally simpler than that available as described by the previous hydrodynamic experts. It was agreed that hydrodynamic and structural analysis code integration is a high priority of the SLSR project and means for achieving this integration were identified.

Finally, experts in reliability, virtual reality, and simulation (UofC, NRC, and UofM) gave examples of how the product of the SLSR program could fit into a larger computer environment where simulation-based designs incorporating probabilistic methods would be possible.

In summary, the workshop was one of the few times where researchers of the disparate disciplines were brought together to develop a coordinated program for ship structural design. Through the spirited discussions, a new awareness of the problems facing the different fields was formed and in this respect, the workshop must be considered a success.

Prof. Armin W. Troesch, Project Director
July 14, 1994

ONR WORKSHOP ON NONLINEAR SEA LOADS AND SHIP RESPONSE: A BASIS FOR SHIP STRUCTURAL DESIGN

College of Engineering
University of Michigan, Ann Arbor

July 7 & 8, 1994

Boulevard Room, North Campus Commons

AGENDA

Thursday, July 7

8:00 - 8:30 Coffee and doughnuts, registration

Workshop Introduction

8:30 - 8:45 Welcome
Prof. Michael G. Parsons, Naval Architecture and Marine Engineering, Associate Dean, College of Engineering, University of Michigan

8:45 - 9:00 Workshop Focus
Dr. Peter Majumdar, Office of Naval Research

9:00 - 9:15 *NAVSEA Initiative in Wave Loads Predictions*
Mr. Allen H. Engle, Naval Sea Systems Command

Hydrodynamics

9:15 - 9:50 *Large-Amplitude Motion and Wave-Load Predictions for Ship Design Assessment*
Dr. Nils Salvesen, SAIC

9:50 - 10:25 *Nonlinear Ship Motions*
Prof. Paul D. Sclavounos, Ocean Engineering, Massachusetts Institute of Technology

10:25 - 10:40 Break

10:40 - 11:15 *Prediction of Nonlinear Loading of Flared Bodies Using a Numerical Towing Tank*
Dr. Brian Maskew, Analytical Methods, Inc.

11:15 - 11:50 *Fully Nonlinear Hydrodynamic Loads Using De-Singularized Methods*
Prof. Robert F. Beck, Naval Architecture and Marine Engineering, University of Michigan

11:50 - 12:25 *Loads Associated With the Hydrodynamic Impact of Flat Wedges*
Prof. William S. Vorus, Naval Architecture and Marine Engineering, University of Michigan

12:25 - 1:25 Lunch

1:25 - 2:00	*Nor[illegible] Hydrodynamic Forces on High Speed Vessels* Pro. Armin W. Troesch, Naval Architecture and Marine Engineering, University of Michigan

Structures and Design

2:00 - 2:35	*Ship Structures and NAVSEA* Mr. Jerome P. Sikora, CDNSWC
2:35 - 3:10	*Integrated Ship Structural Design Methodology* Mr. Tobin R. McNatt, Ross and McNatt and Prof. Owen Hughes, Aerospace and Ocean Engineering, Virginia Polytechnic Institute & State University
3:10 - 3:25	Break
3:25 - 4:00	*Probabilistic Loading of Ship Structures by Slamming* Prof. William Webster, Naval Architecture and Offshore Engineering, University of California, Berkeley (for Prof. Alaa Mansour)
4:00 - 4:35	*Dynamic Loading Approach for Analyzing the Ship Structure* Dr. Yung-Sup Shin, American Bureau of Shipping

Friday, July 8

8:00 - 8:30	Coffee and doughnuts

Simulation-Based Design Environment

8:30 - 8:50	*Use of Reliability in Structural Design* Mr. Robert A. Sielski, Marine Board, National Research Council
8:50 - 9:25	*Virtual Reality in Design and Manufacturing* Prof. K.-Peter Beier, Naval Architecture and Marine Engineering, University of Michigan
9:25 - 10:00	*The Role of Simulation in Ship Design: Some Cautionary Examples* Prof. Armin W. Troesch, Naval Architecture and Marine Engineering, University of Michigan
10:00 - 10:15	Break
10:15 - 10:50	Continued discussion: *Dynamic Loading Approach for Analyzing the Ship Structure* Dr. Yung-Sup Shin, American Bureau of Shipping

Workshop Wrap-up

11:25 - 12:00	Dr. Peter Majumdar, Office of Naval Research

ONR WORKSHOP ON NONLINEAR SEA LOADS AND SHIP RESPONSE: A BASIS FOR SHIP STRUCTURAL DESIGN
JULY 7-8, 1994

LIST OF ATTENDEES

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ONR WORKSHOP ON NONLINEAR SEA LOADS AND SHIP RESPONSE: A BASIS FOR SHIP STRUCTURAL DESIGN
JULY 7-8, 1994

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HYDRODYNAMIC LOADS TECHNOLOGY DEVELOPMENT PROGRAM (A STATUS REPORT)

NAVAL SEA SYSTEMS COMMAND

ALLEN ENGLE
HYDRODYNAMICS DIVISION
NAVAL SEA SYSTEMS COMMAND
(703) 602-9297

OUTLINE

- PROGRAM OBJECTIVES
- LOADS PREDICTION APPROACH
- MAJOR EFFORTS TO DATE
  - FULL SCALE TRIALS
  - MODEL TESTS
  - ANALYTIC TOOL DEVELOPMENT
- COOPERATIVE RESEARCH W/NORWAY

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PROGRAM OBJECTIVES

- **IMPROVE OUR ABILITY TO PREDICT HYDRODYNAMIC LOADS IN ORDER TO:**
  - ESTABLISH A BASIS FOR NEW DESIGN PROCEDURES (INCLUDING RELIABILITY BASED STRUCTURAL DESIGN)
  - ESTABLISH A BASIS FOR STRUCTURAL DESIGN CONCEPTS WHICH ARE OUTSIDE THE HISTORIC DATA BASE
  - FULLY EXPLOIT EXISTING SOPHISTICATED IN-HOUSE STRUCTURAL ANALYSIS TECHNIQUES
  - MAXIMIZE EFFICIENCY OF STRUCTURE, REDUCE WEIGHT AND MAINTENANCE

SAFE, RELIABLE, AFFORDABLE, INNOVATIVE DESIGNS

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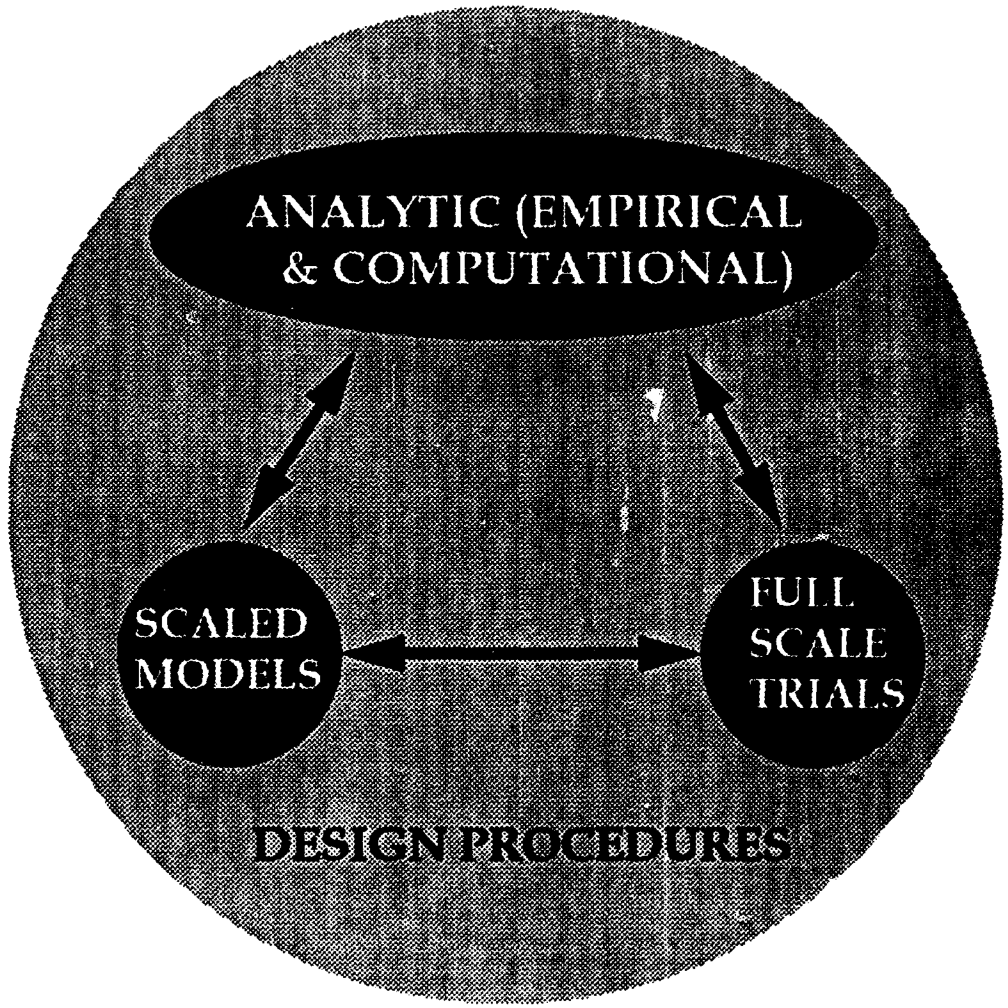
LOADS PREDICTION APPROACH



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MAJOR EFFORTS TO DATE

- HYDRODYNAMIC LOADS DATA COLLECTED FOR THE FOLLOWING:
  - CG 47 CLASS
    - FULL SCALE TRIALS
    - MODEL TESTS
  - LHD 1 CLASS
    - FULL SCALE TRIALS (SHIP INSTRUMENTED, AS OF YET NO DATA COLLECTED)
    - MODEL TESTS
  - HMAS SWAN (AUSTRALIAN SHIP)
    - FULL SCALE TRIALS
  - CPF (CANADIAN PATROL FRIGATE)
    - MODEL TESTS
    - FULL SCALE TRIALS (PLANNED)

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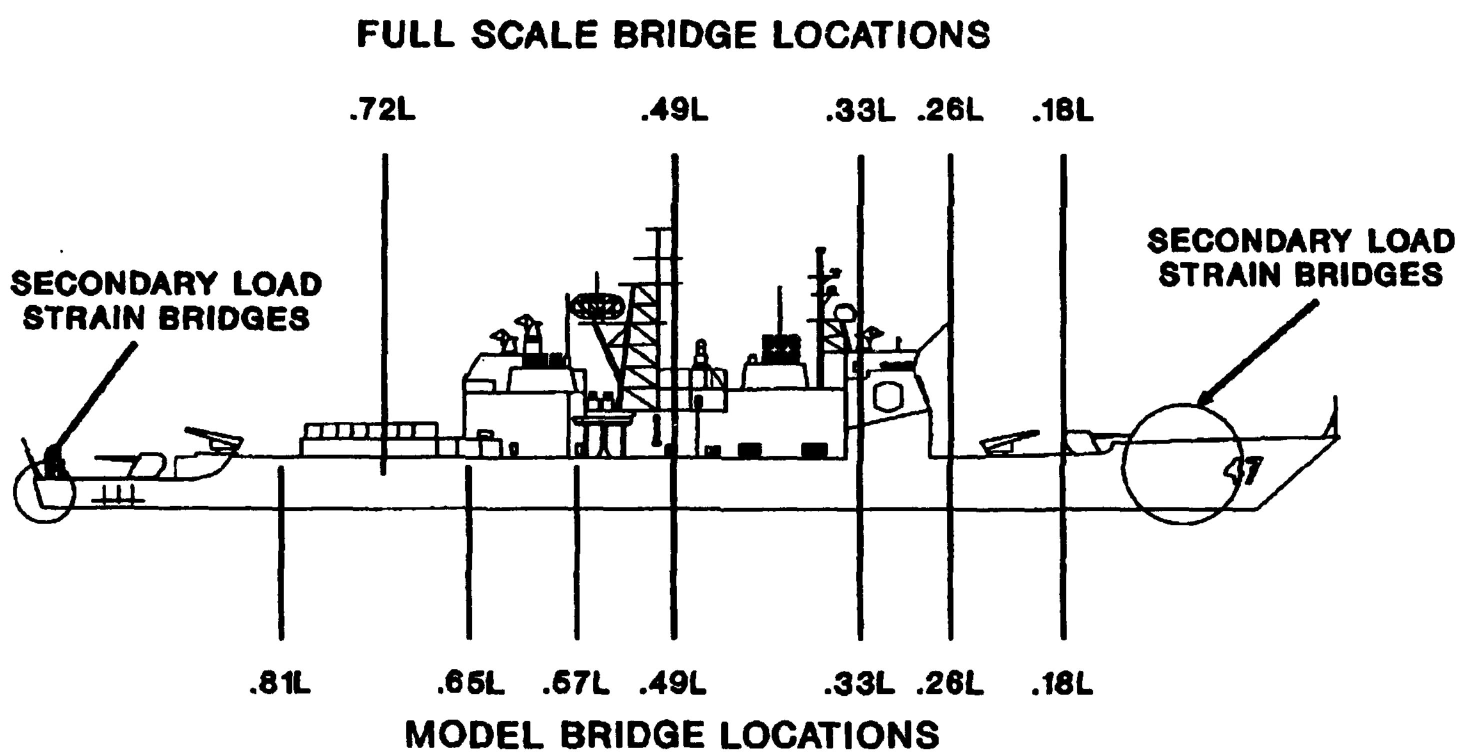
STRAIN GAGE LOCATIONS



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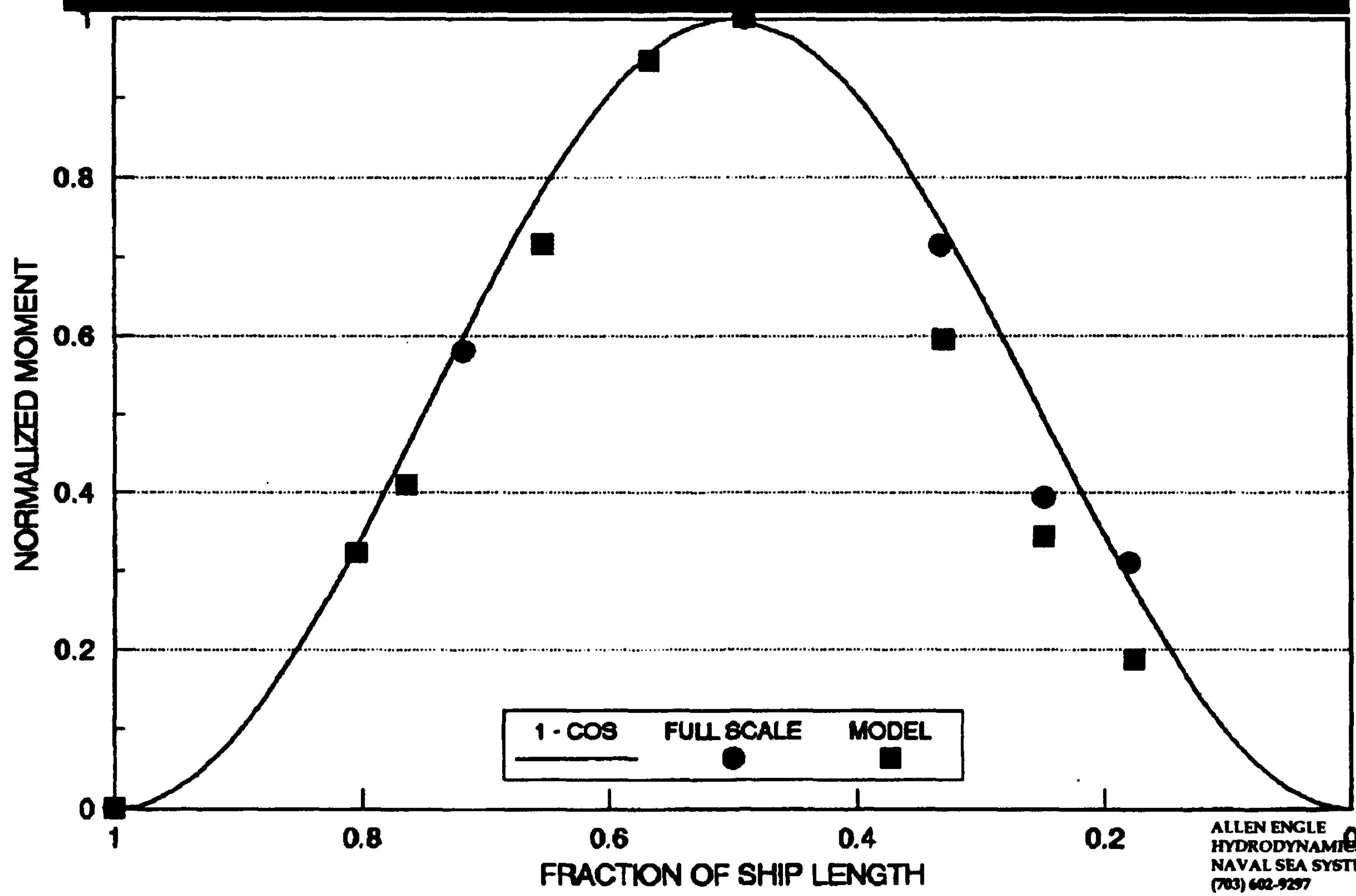
BENDING MOMENT DISTRIBUTION (ORDINARY WAVE - VERTICAL)



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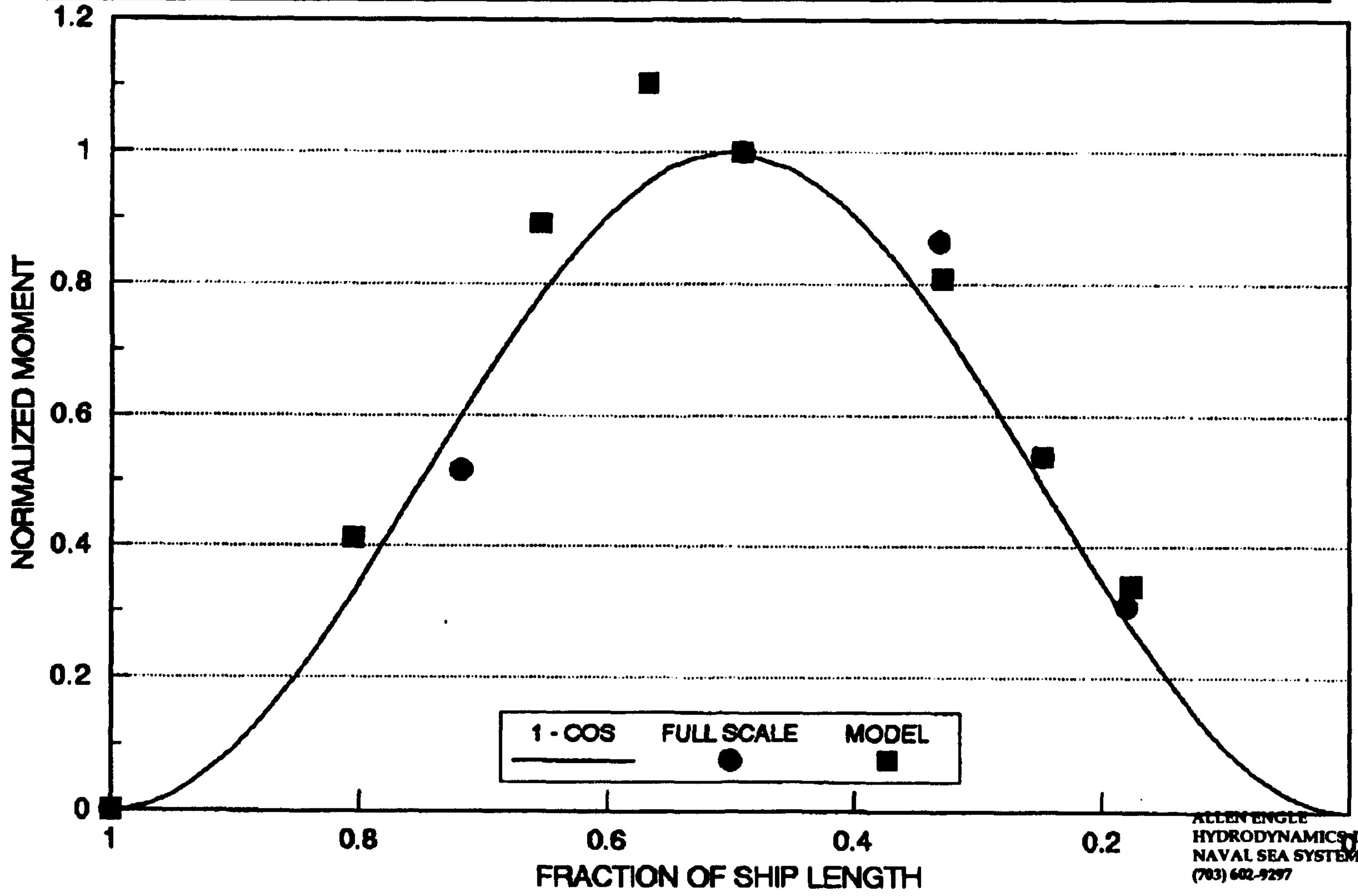
BENDING MOMENT DISTRIBUTION (ORDINARY WAVE - LATERAL)



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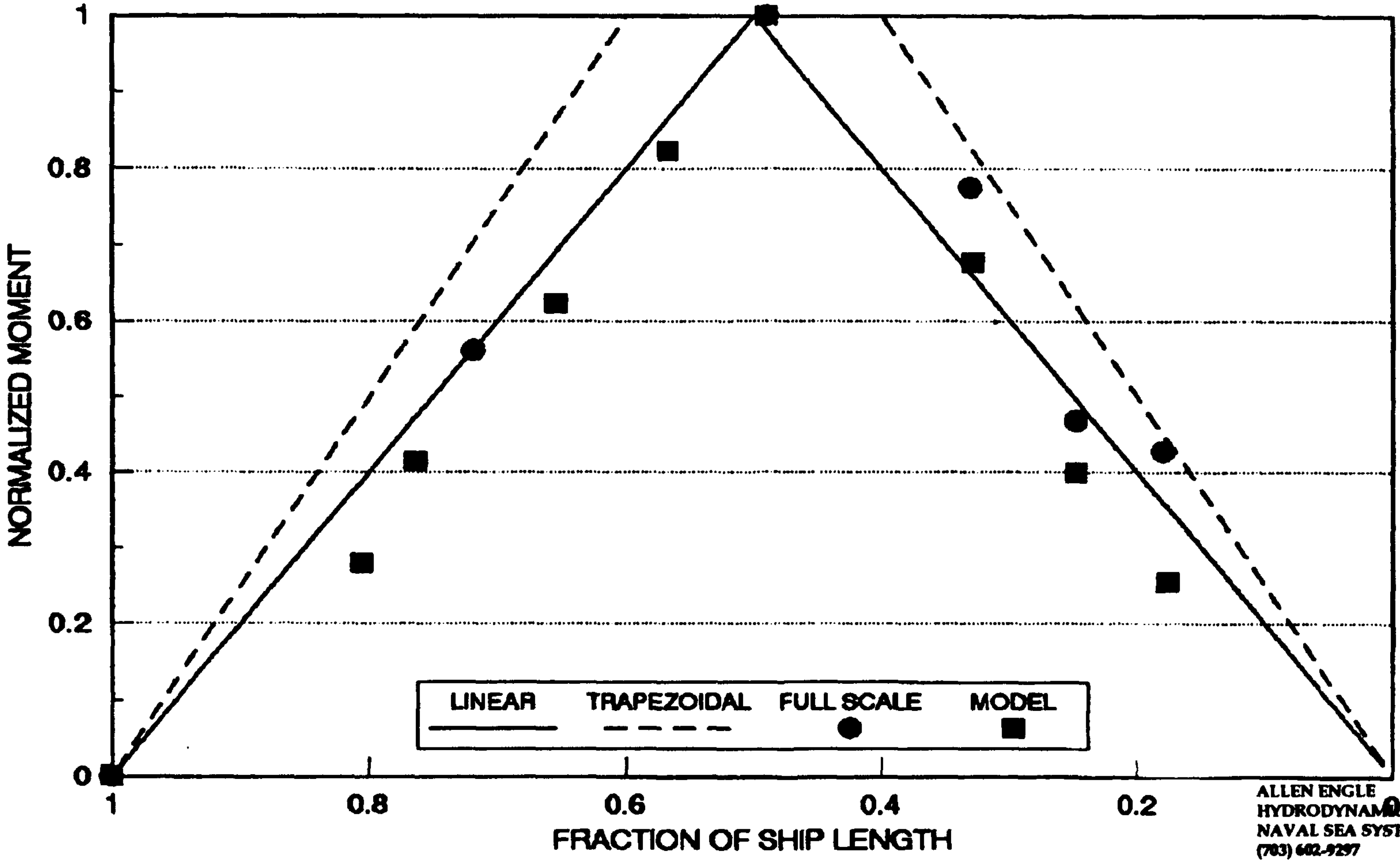
BENDING MOMENT DISTRIBUTION (VERTICAL WHIPPING ONLY)



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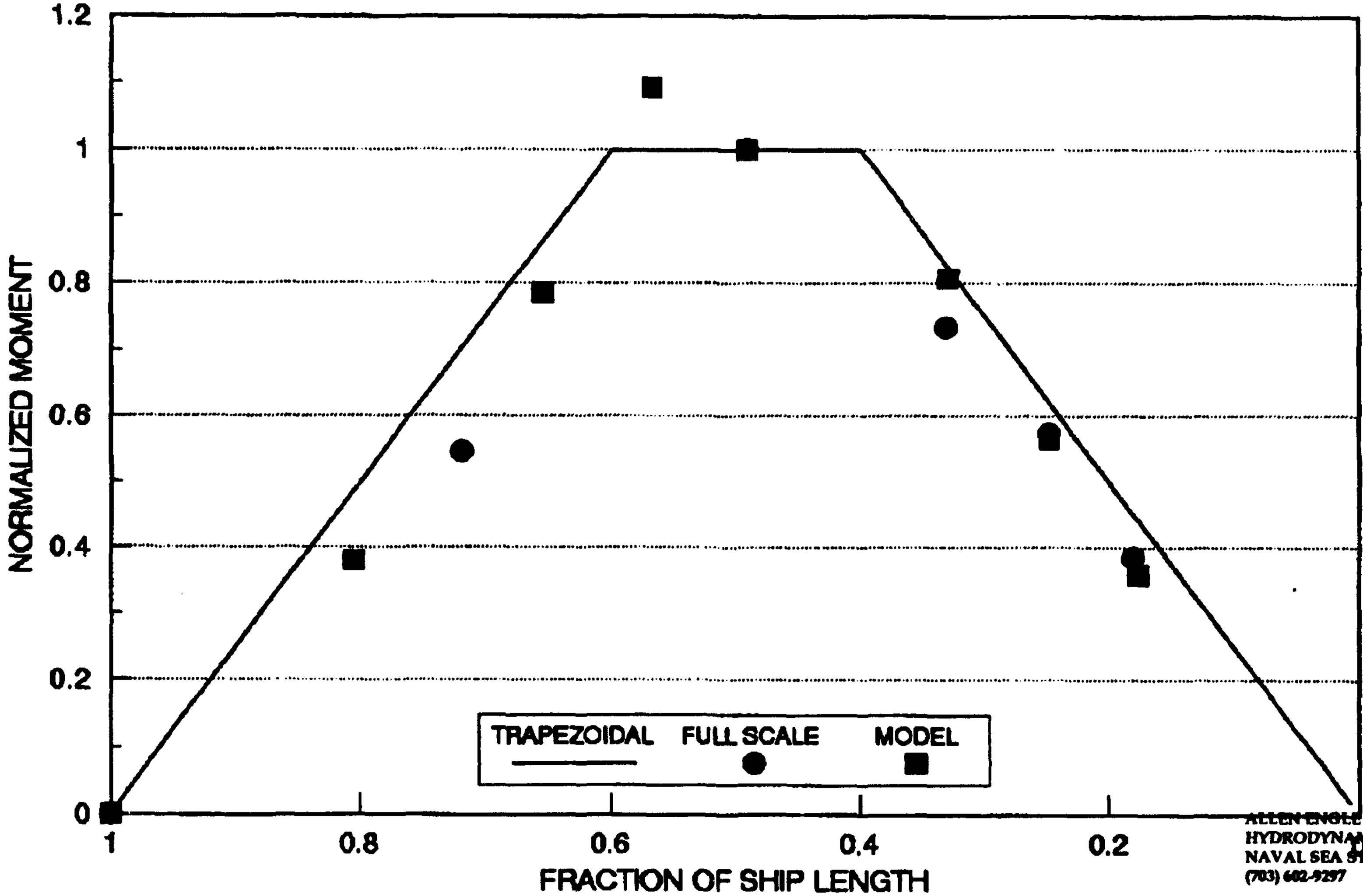
BENDING MOMENT DISTRIBUTION (LATERAL WHIPPING ONY)



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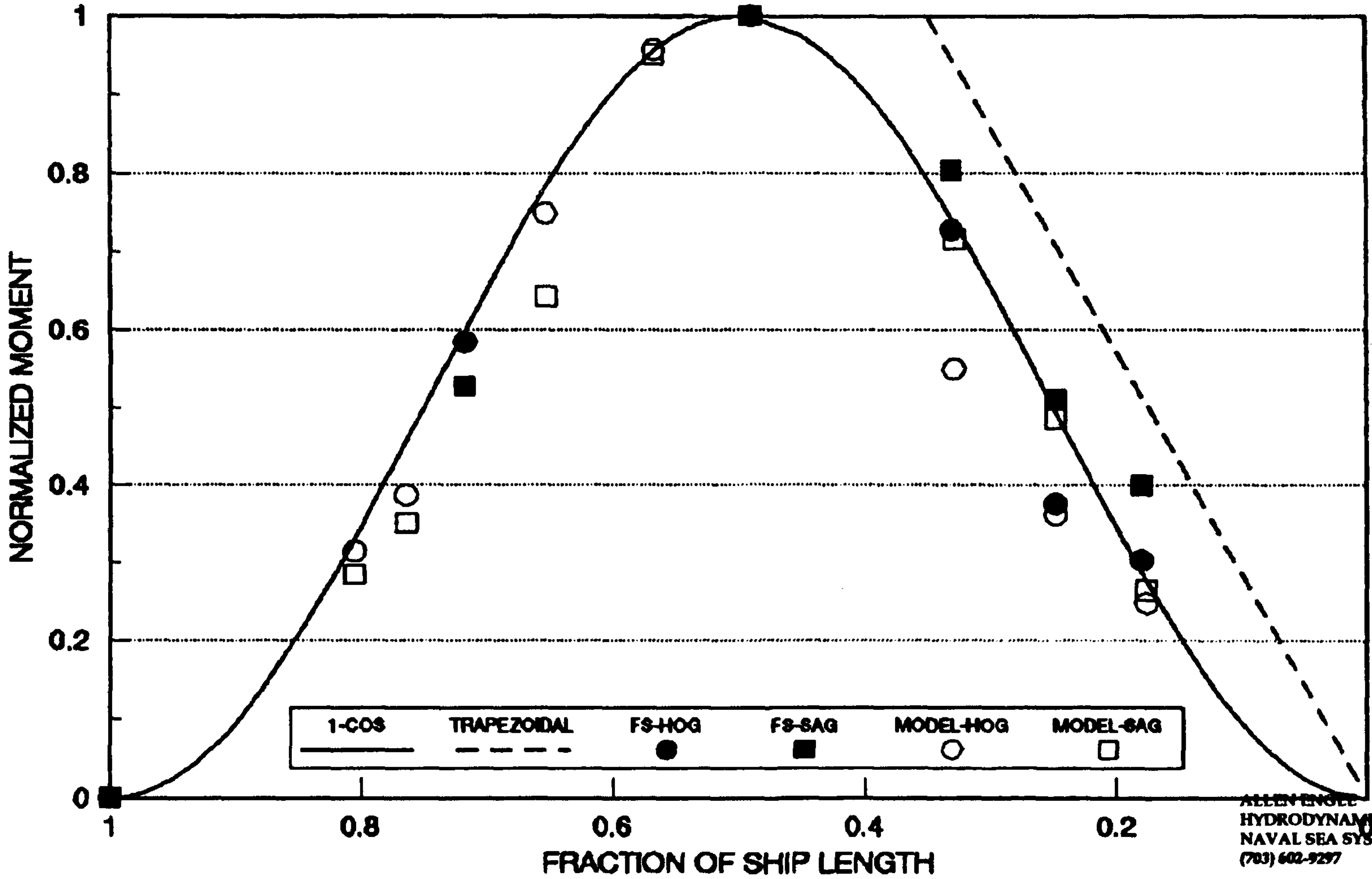
BENDING MOMENT DISTRIBUTION (WAVE + WHIPPING - VERTICAL)



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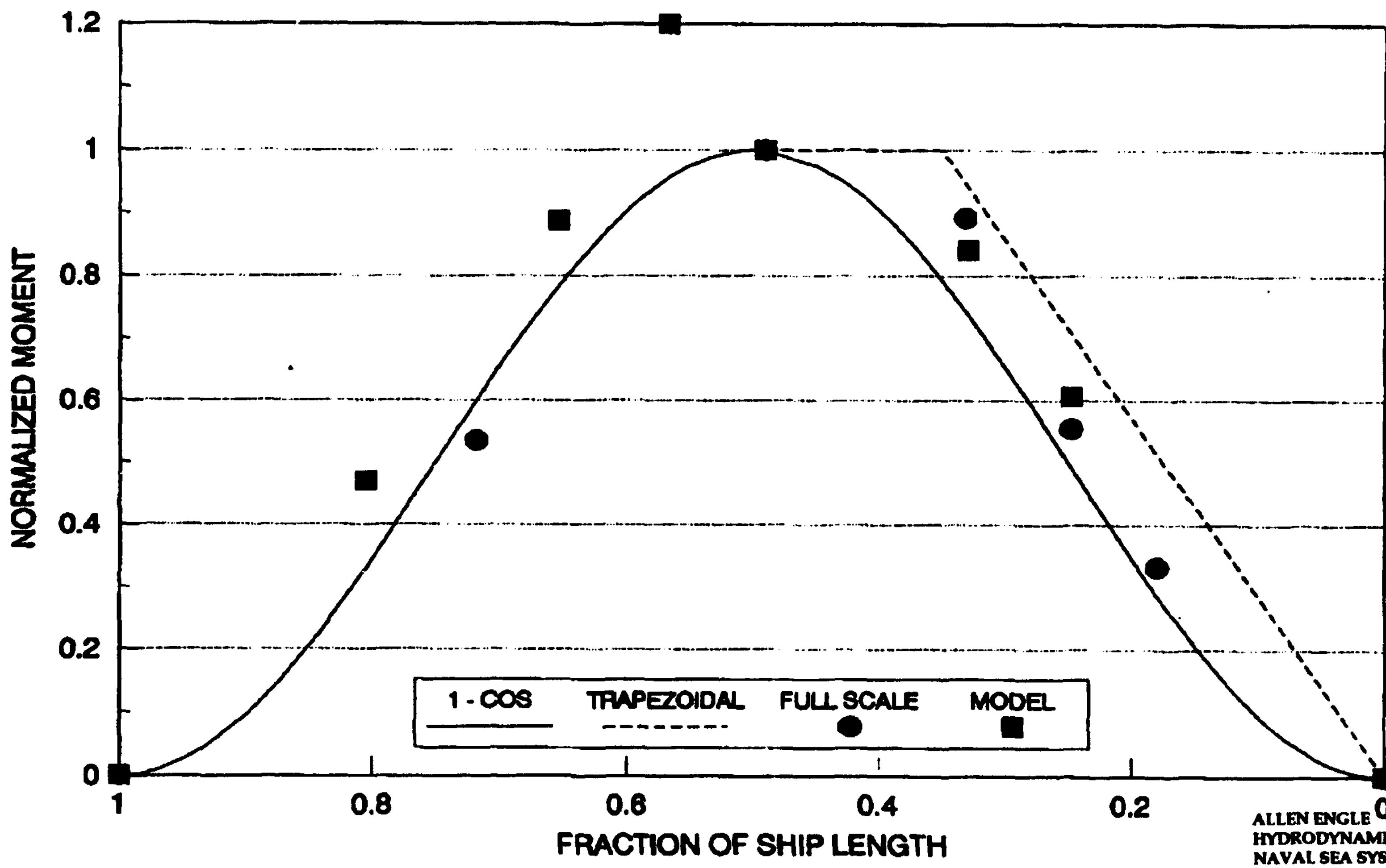
BENDING MOMENT DISTRIBUTION (WAVE + WHIPPING - LATERAL)



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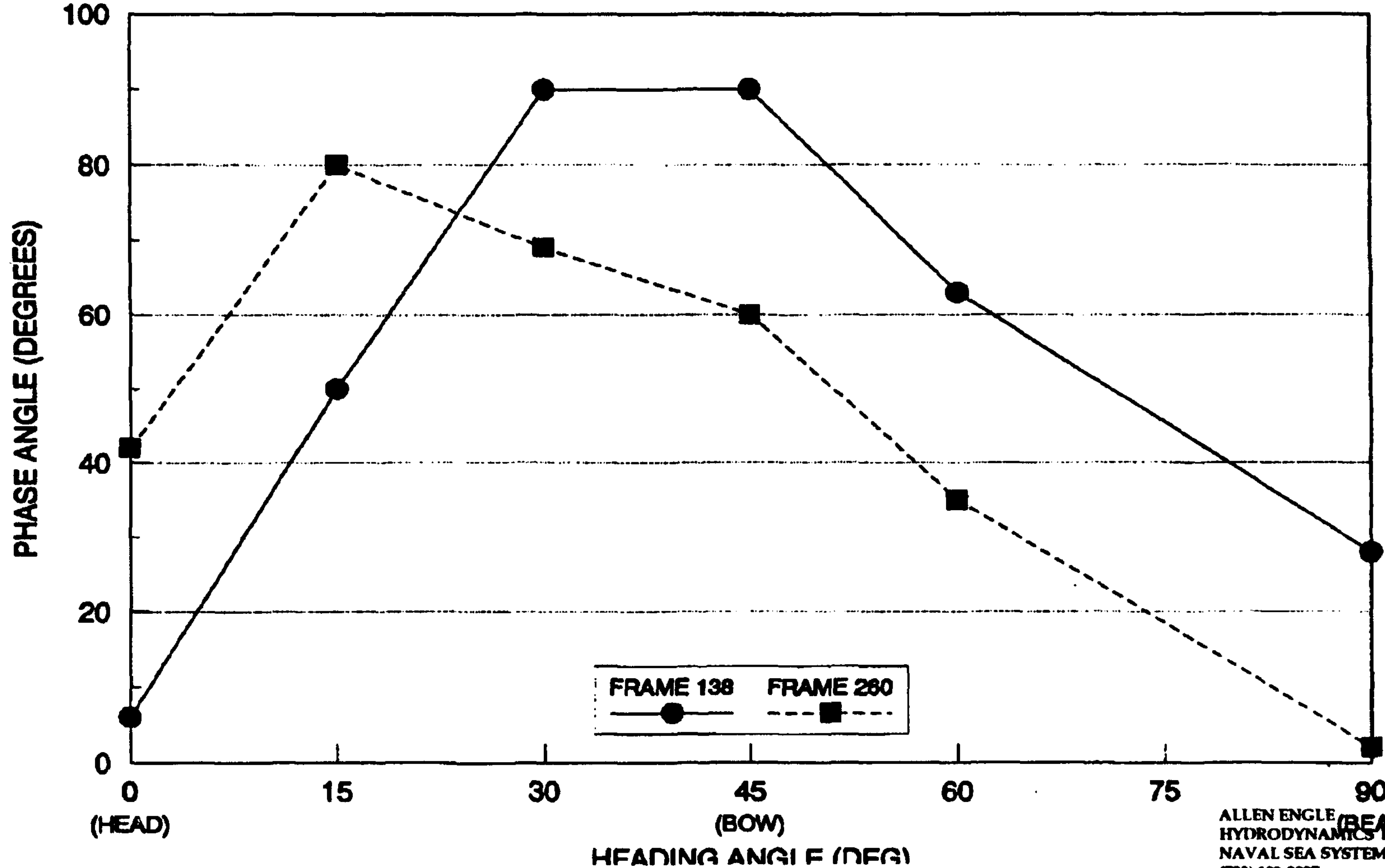
PHASE ANGLE BETWEEN VERTICAL AND LATERAL BENDING MOMENTS



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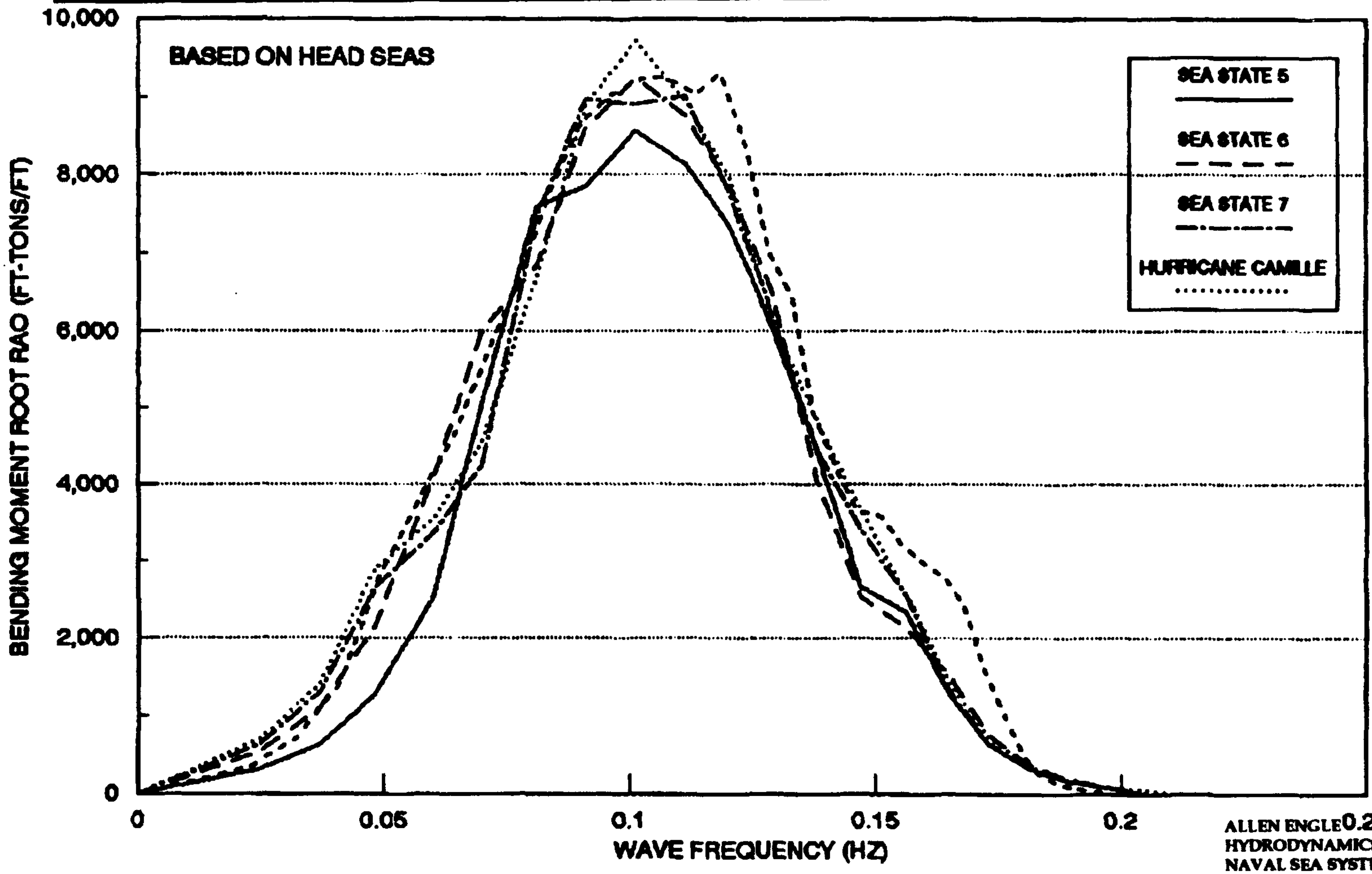
VERTICAL BENDING MOMENT RAO'S FUNCTION OF SEA STATE



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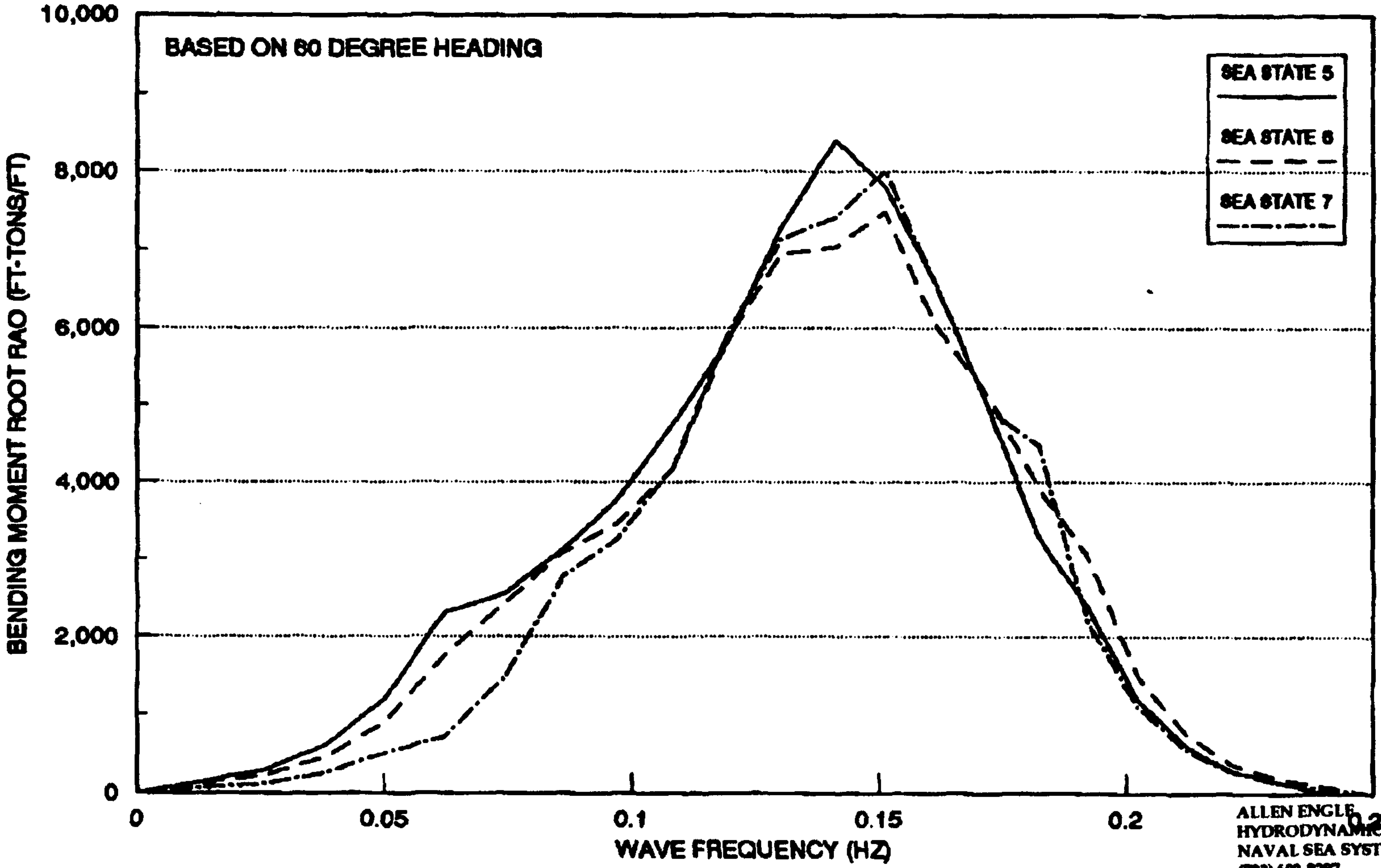
LATERAL BENDING MOMENT RAO'S FUNCTION OF SEA STATE



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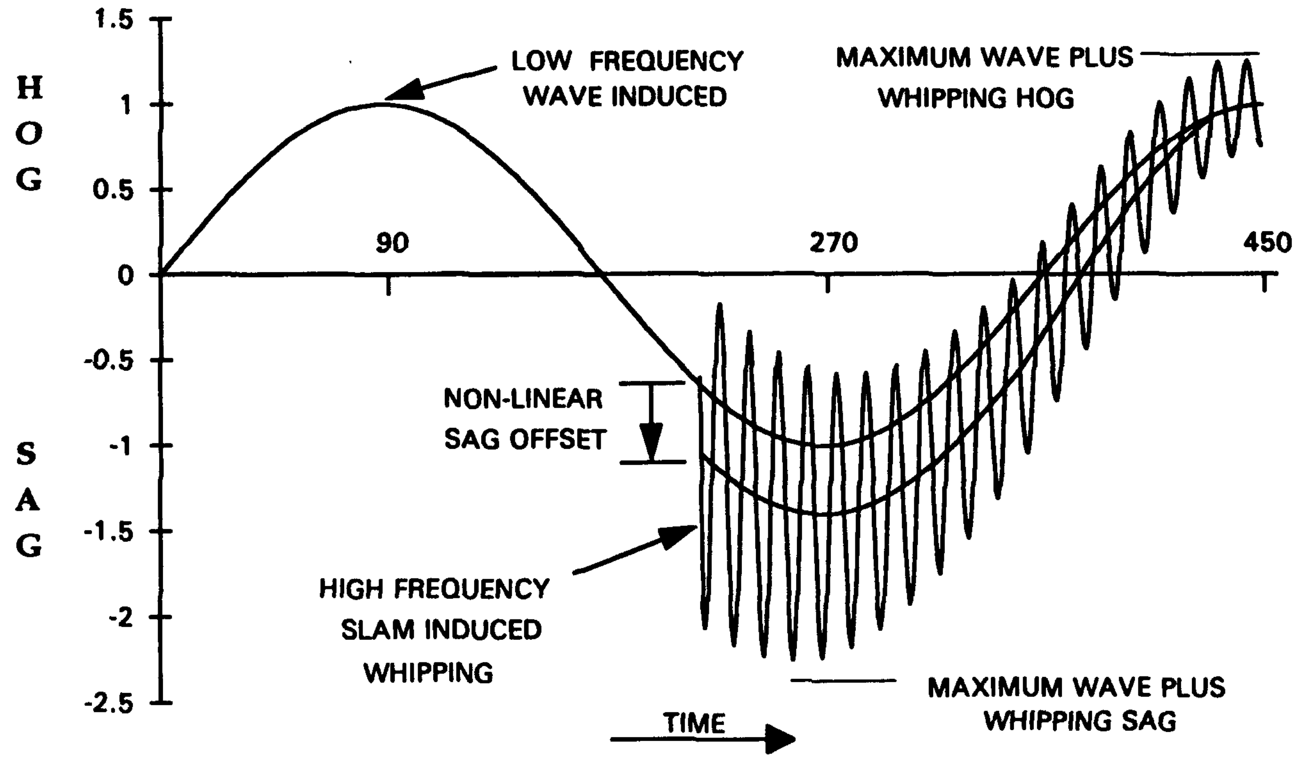
PHASE ANGLE BETWEEN HULL GIRDER BENDING DUE TO VERTICAL ORDINARY WAVE AND WHIPPING



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HIGHLIGHTS OF CG 47 CLASS TEST DATA

■ MAJOR FINDINGS:

- COMBINED LATERAL WAVE INDUCED PLUS WHIPPING LOAD
  - <u>SAME ORDER OF MAGNITUDE</u> AS VERTICAL WAVE INDUCED PLUS WHIPPING LOAD
- VERTICAL AND LATERAL WHIPPING MOMENTS:
  - TRAPEZOIDAL DISTRIBUTION PROVIDES BEST FIT
    - CONSTANT MAXIMUM AMPLITUDE = 40 TO 60% SHIP LENGTH
- ONSET OF WHIPPING WILL OCCUR:
  - VERTICAL BENDING: 220 DEGREES INTO THE ORDINARY WAVE CYCLE
  - LATERAL BENDING: 248 DEGREES INTO THE ORDINARY WAVE CYCLE

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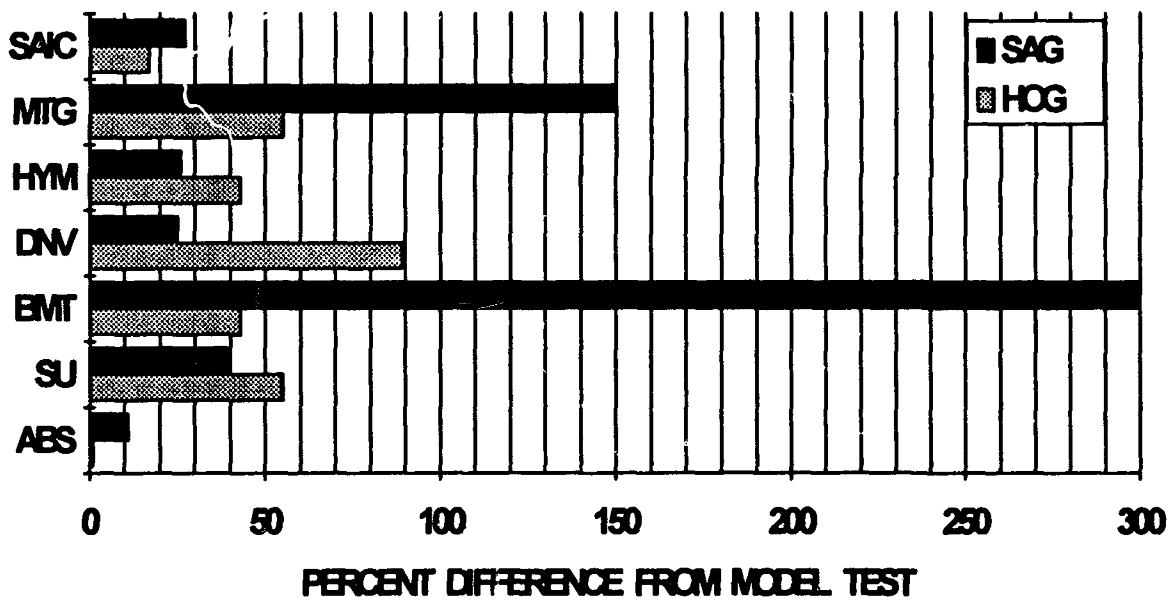
HYDRODYNAMIC LOAD PREDICTIONS

COMPARISON OF
THE STATE OF THE ART



CG 47 HULL FORM

SHIP SPEED = 10 KNOTS

SEA STATE = HURRICANE CAMILLE

NOTE:
1. ABS CODE IS NOT A TRUE TIME DOMAIN CODE
2. SAIC CODE COMPUTATION BASED UPON SCALED DOWN WAVE HEIGHT AND DOES NOT INCLUDE SLAM INDUCED WHIPPING EFFECTS

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HYDRODYNAMIC LOAD PREDICTIONS

- BASED ON "SHOOT-OUT" RESULTS:
  - ACQUIRED "PRODUCTION" VERSIONS OF:
    - QUASI-LINEAR TIME DOMAIN CODE, HYDROMECHANICS
    - LARGE AMPLITUDE STRIP THEORY CODE, MTG
  - IDENTIFIED NEED TO ACCELERATE EMERGENT NON-LINEAR TIME DOMAIN CODE
    - COOPERATIVE RESEARCH W/NORWAY

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COOPERATIVE RESEARCH EFFORT UNITED STATES & NORWAY

■ DYNAMIC ANALYSIS SUPPORT SYSTEM

- TECHNICAL AREAS
  - WAVE ENVIRONMENT
  - LOADS & MOTIONS COMPUTER CODES
  - STRUCTURAL INTERFACE CODE
  - DESIGN METHODOLOGY
  - HYDROELASTICITY
  - DYNAMIC STABILITY
- 5 YEAR PROGRAM
  - DEVELOP DESIGN TOOLS
    - DEVELOPMENT PERIOD, 3 YEARS
    - DESIGN INTEGRATION/IMPLEMENTATION, 2 YEARS

SOMETHING WORTH DOING IS WORTH DOING "BADLY"

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LARGE-AMPLITUDE MOTION AND WAVE-LOAD PREDICTIONS FOR SHIP DESIGN ASSESSMENT

Presented at

ONR Workshop on Nonlinear Sea Loads
University of Michigan
July 7 and 8, 1994

Presented by

Nils Salvesen
Ship Technology Division
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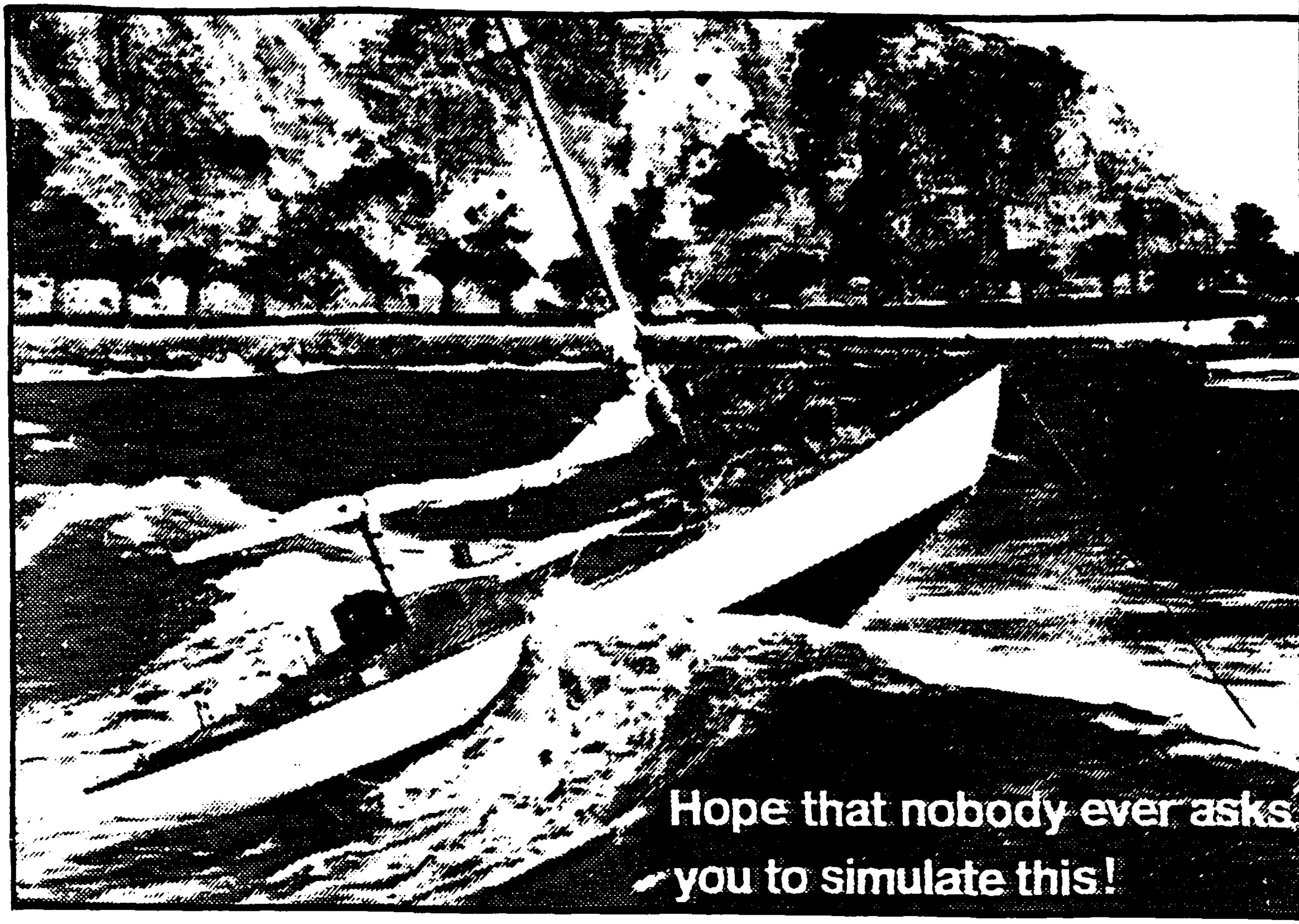
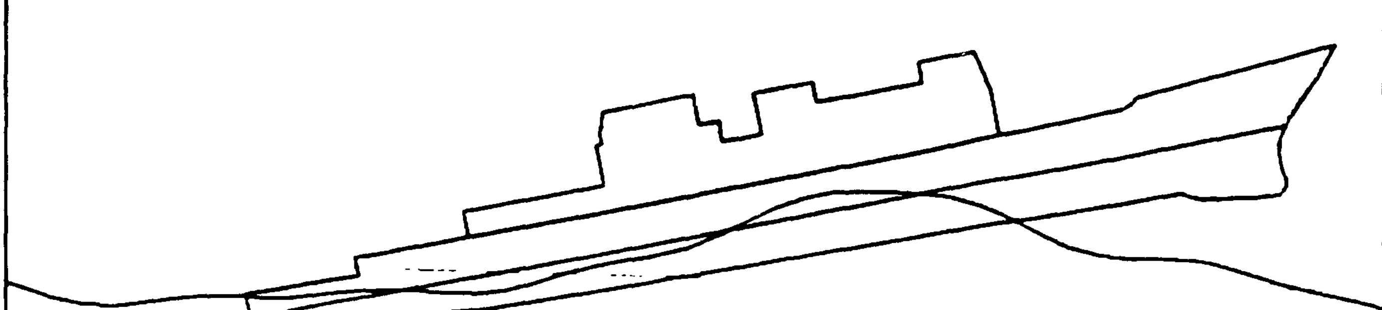
An Employee-Owned Company



This is exactly what we intend to simulate.



MOTION, WAVE-LOAD AND STRUCTURAL RESPONSE PREDICTIONS

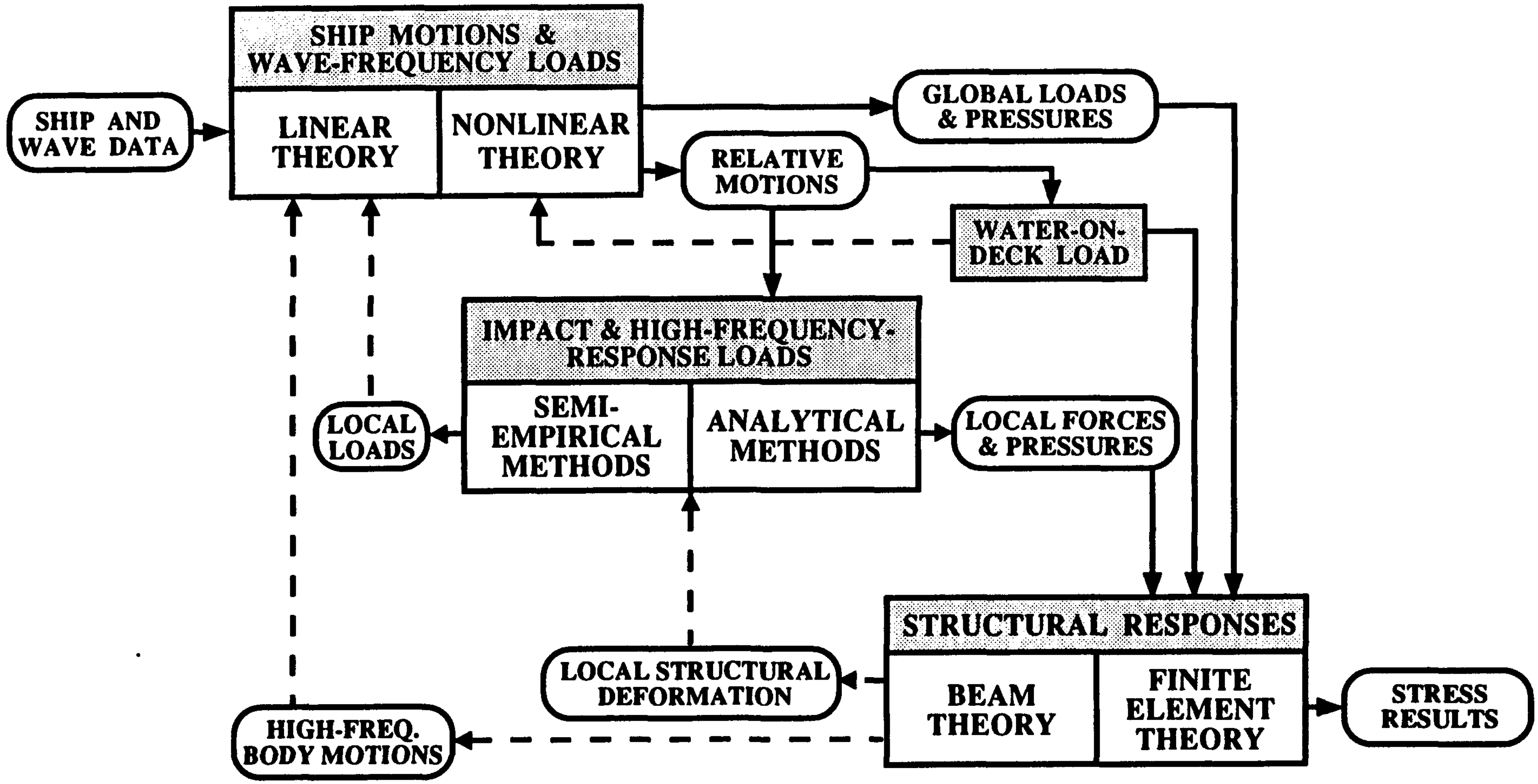
Objective: Develop a Ship Design Assessment System

- Fully Integrated
- Multi-Level Codes
- Robust & Efficient
- Validated (Risk & Uncertainty)
- Available to Community

Major Components:

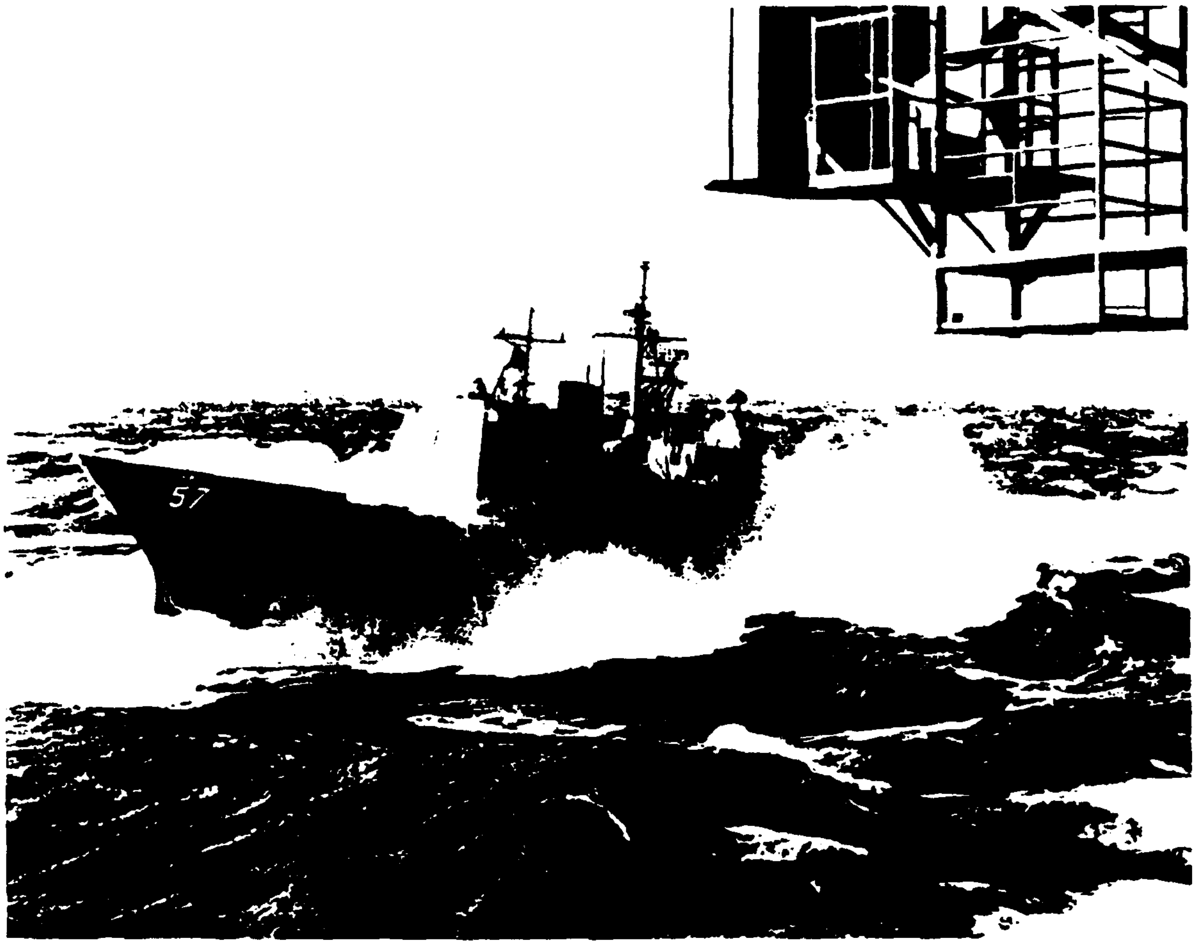
1. Geometry and Ship Definition
2. Wave Events and Sea Conditions
3. Ship Motion Calculations
4. Wave Load Calculations
5. Structural Responses (Stresses)
6. Structural Design Approach

SIMULATION OF SHIP MOTIONS AND STRUCTURAL RESPONSES



MARINTEK SINTEF GROUP

An Employee-Owned Company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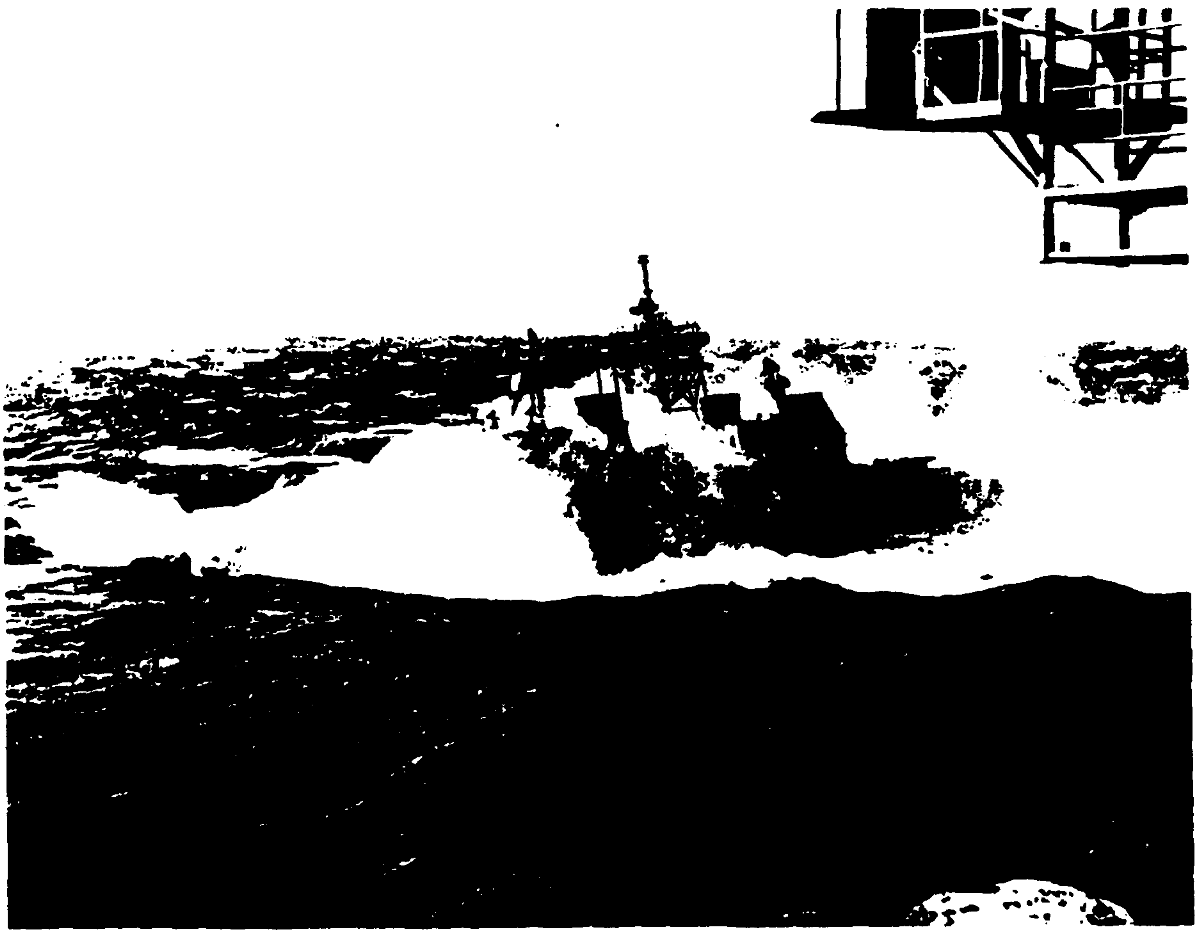
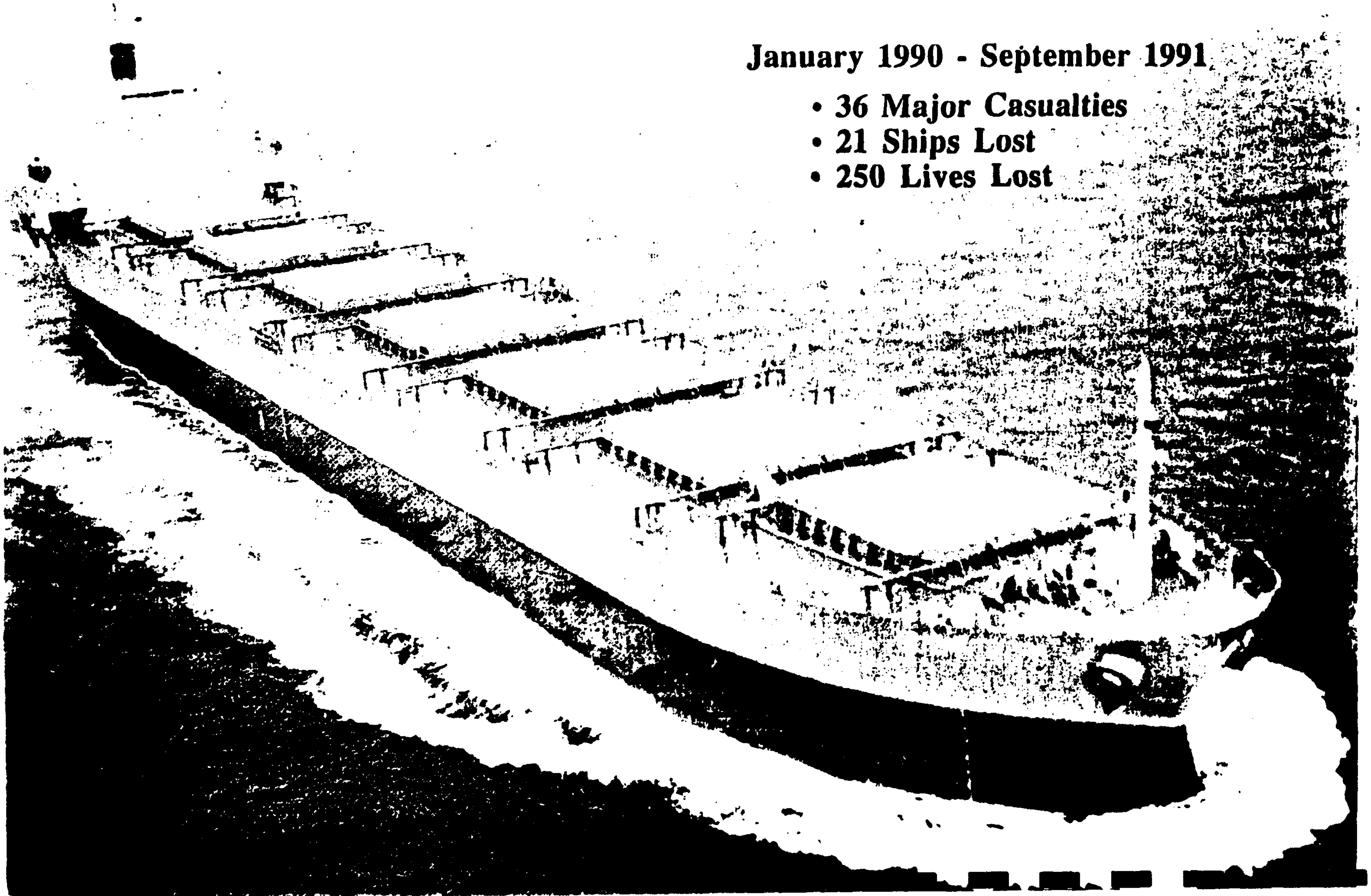


"BULK CARRIERS, A CAUSE FOR CONCERN"

Phil Rynn, ABS/Houston



OUTLINE OF PRESENTATION AND SPONSORS

I. Status Report on IDEAS Motion and Load System

A. System Development

1. ARPA (1988-90)	• System Integration • Large-Amplitude Code
2. USCG (1989-94)	• Safety (IMO) and Capsizing
3. ABS (1992-94)	• Motions and Loads for Structural Design
4. ONR (1992-94)	• LAMP Development • Installation at Tech Center

B. System Applications

1. NAVSEA (1990-91)	• CG47 AEGIS Calculations
2. ARPA (1993-94)	• Simulation Based Design
3. ARPA (1994-97)	• Hypercomputing and Design (Rutgers)

C. Related Work

1. ONR/USCG (1992-94)	• High-Speed Craft with MARINTEK
2. ONR (1990-94)	• Nonlinear Seakeeping RANS/ Potential Flow Coupling

II. Needed Improvements and Extensions

LAMP CODE FOMULATION

1. Potential Flow — Viscous Roll Damping included

2. Body Motions and Incident Waves can be large relative to ship's draft.

3. Body Boundary Condition satisfied on instantaneous wetted surface below incident wave profile.

4. Free-Surface Condition linearized about the incident wave surface — assuming that diffracted waves are small.

LAMP CODE FORMULATION

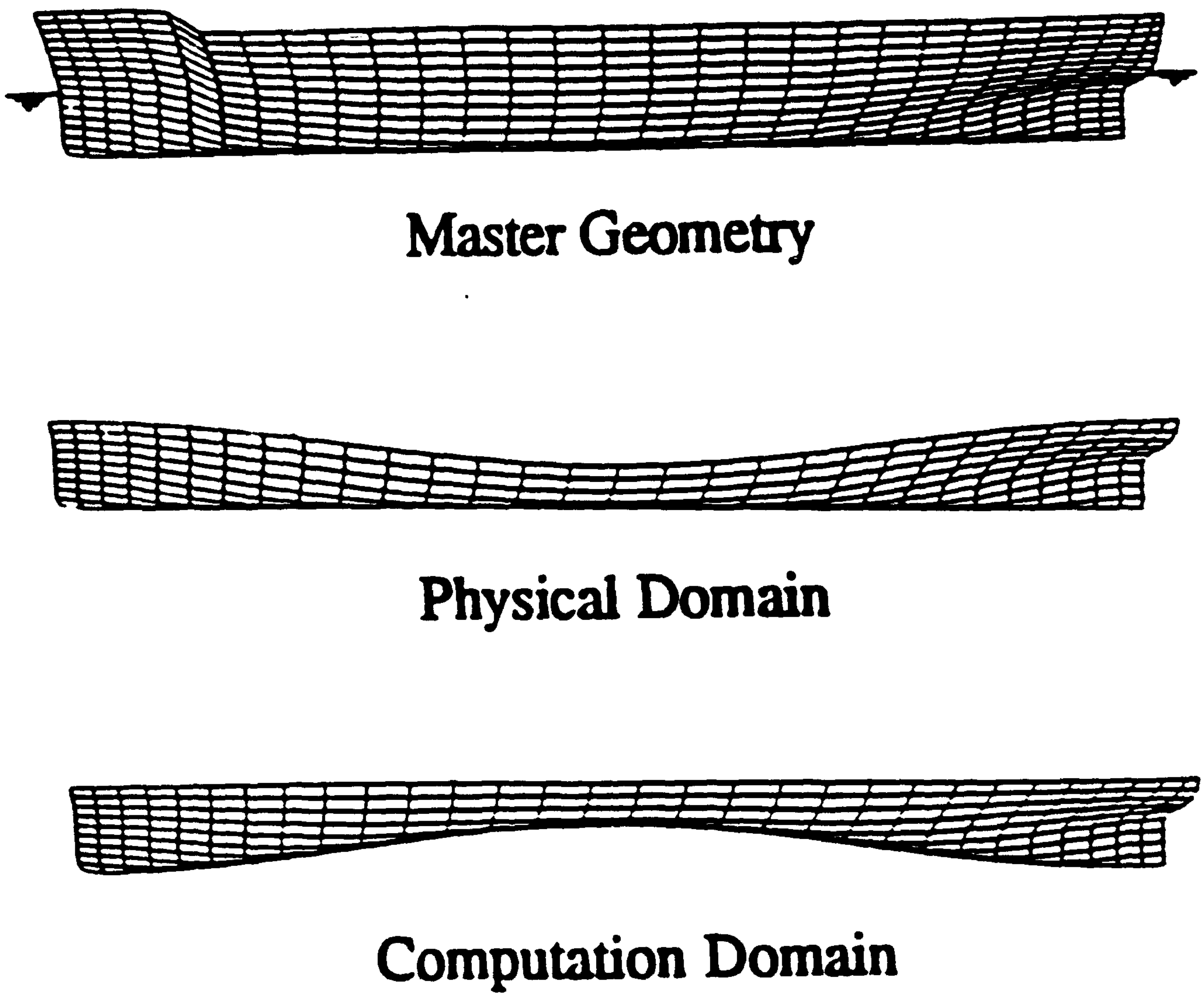


Figure 2: Master Geometry and Panel Distribution in both Physical and Computation Domains.

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LAMP MULTI-LEVEL CODE SYSTEM FOR SHIP MOTIONS AND WAVE LOADS

LAMP-4: The large-amplitude 3-D nonlinear method

LAMP-3: The large-amplitude 2-1/2-D nonlinear method

LAMP-2: The approximate large-amplitude 3-D nonlinear method

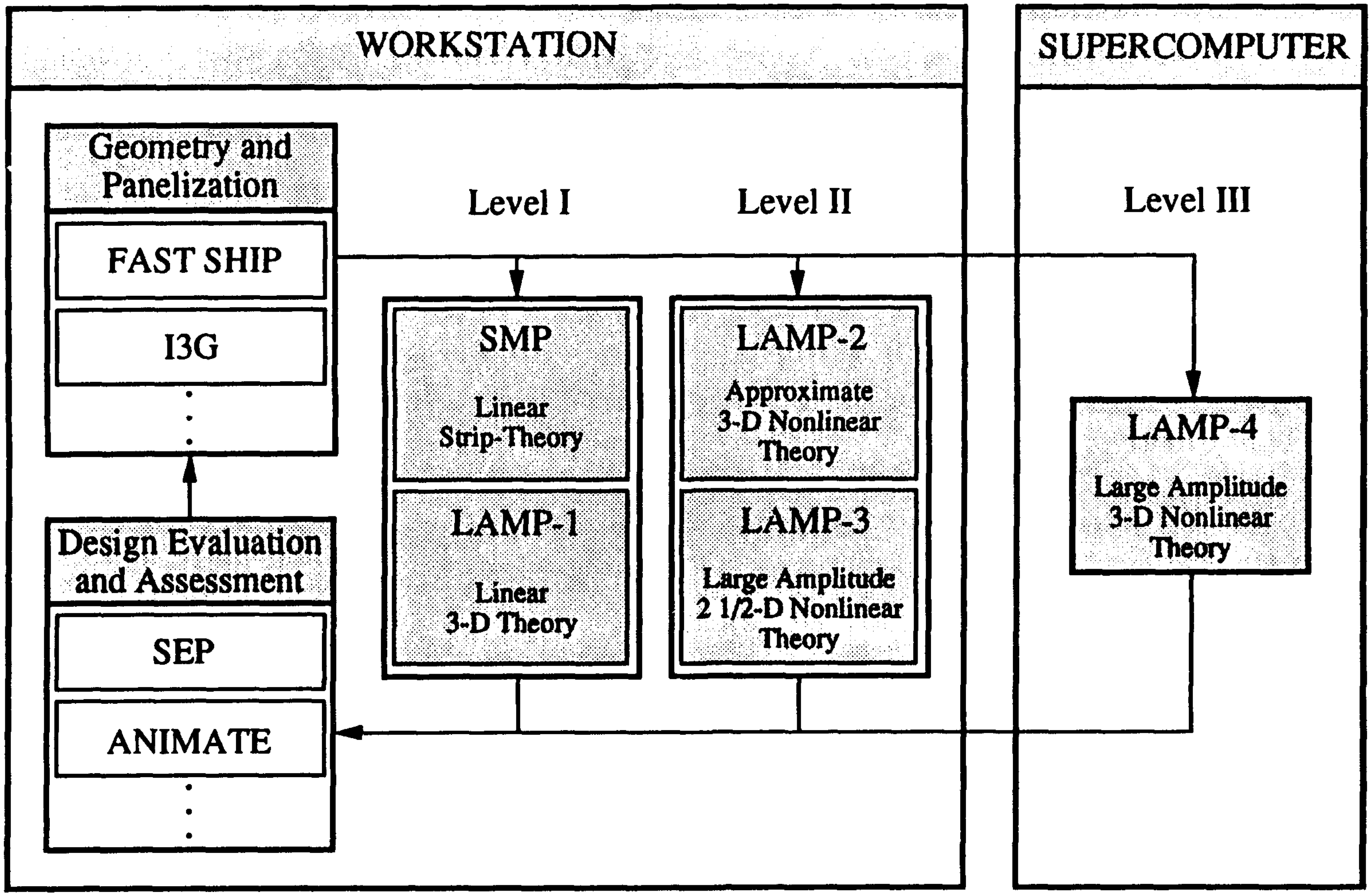
LAMP-1: The linearized 3-D time-domain method

SMP: The U.S. Navy linear strip-theory Ship Motion Program

CPU Time Requirement

	IBM RS6000/550 Workstation	CRAY-YMP Supercomputer
SMP	2.5 seconds	0.5 seconds
LAMP-1	5.0 minutes	1.0 minutes
LAMP-2	6.0 minute	1.2 minute
LAMP-3	-	-
LAMP-4	4.0 hours	0.8 hours

The Present IDEAS Ship Motion and Wave Load System



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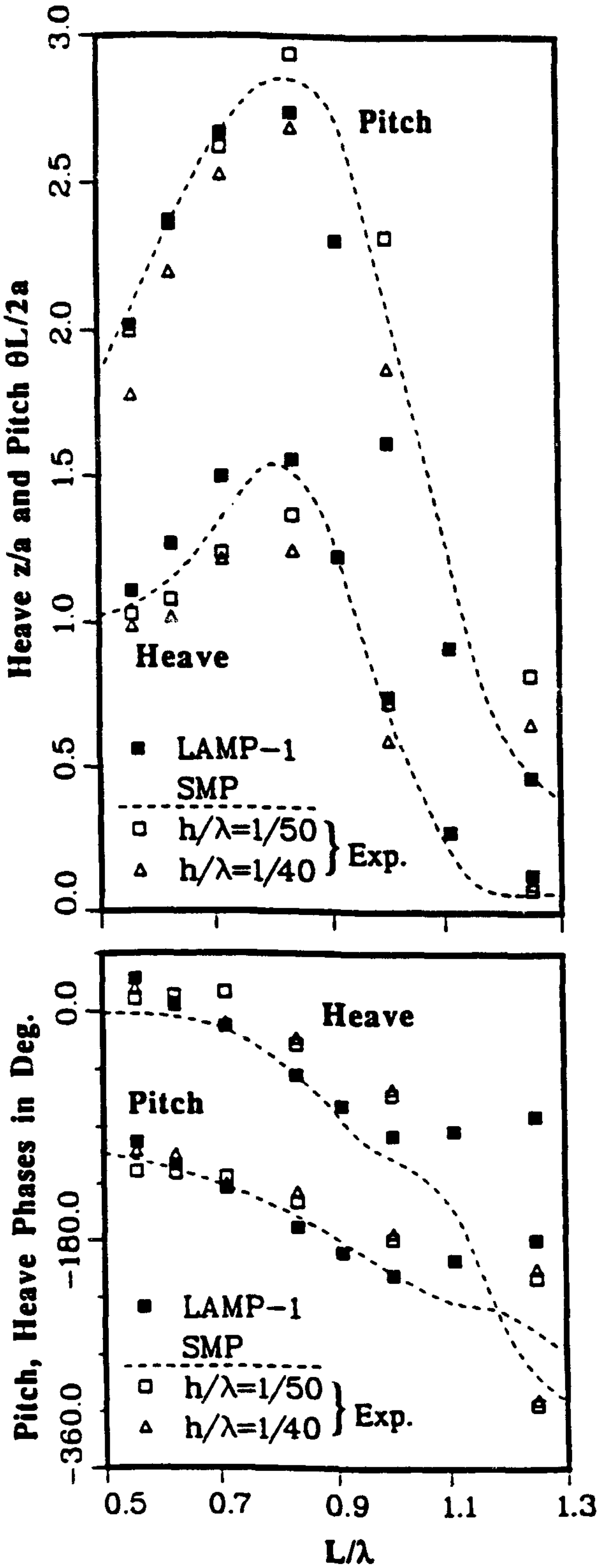
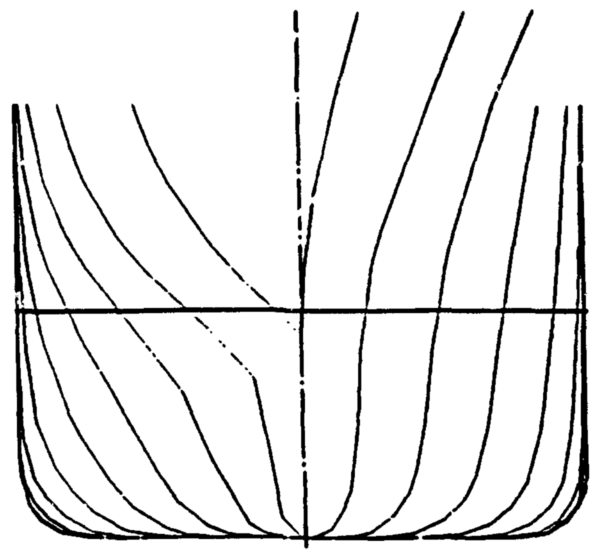
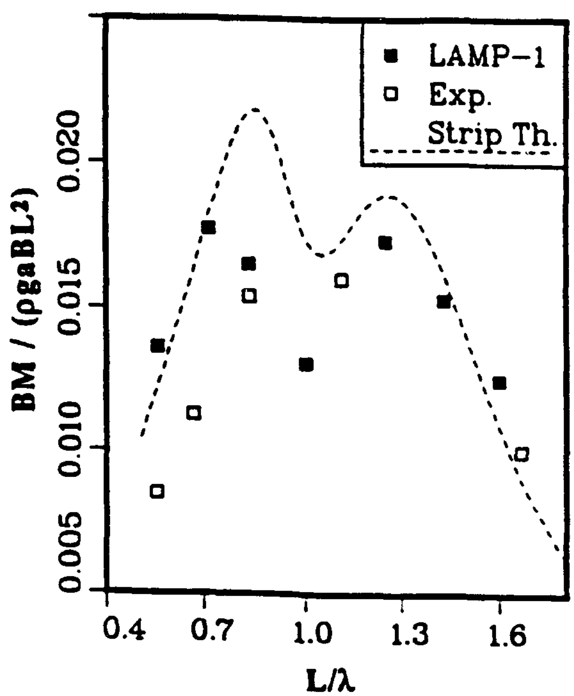
VALIDATION — SERIES 60





Series 60. $C_B = 0.70$



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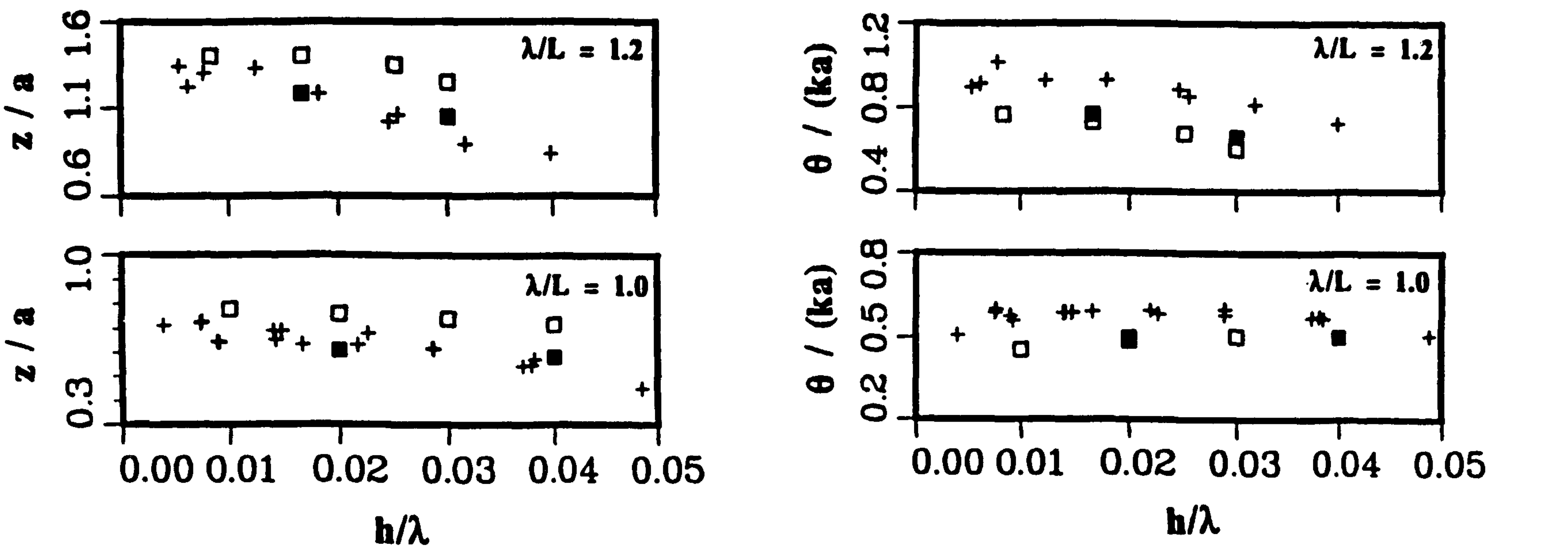
LAMP VALIDATION — S175 CONTAINERSHIP



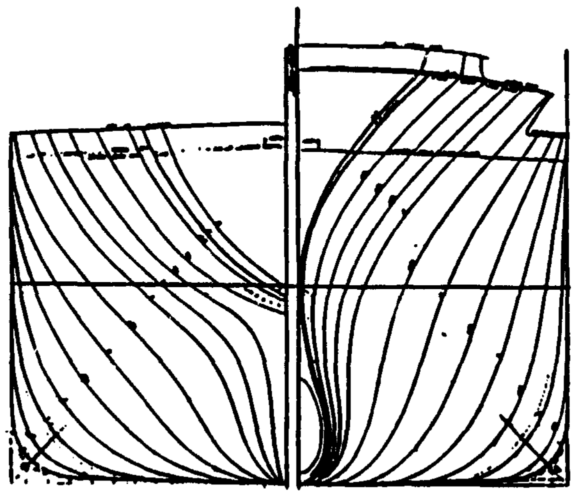


Figure 7: Comparison of Nonlinear Theories (LAMP-2, □, and LAMP-4, ■) and Experiments (+).

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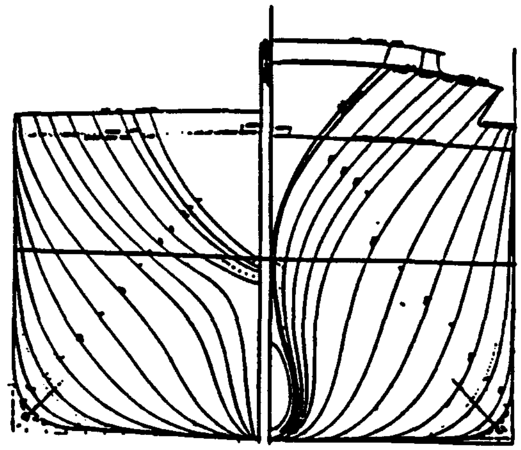
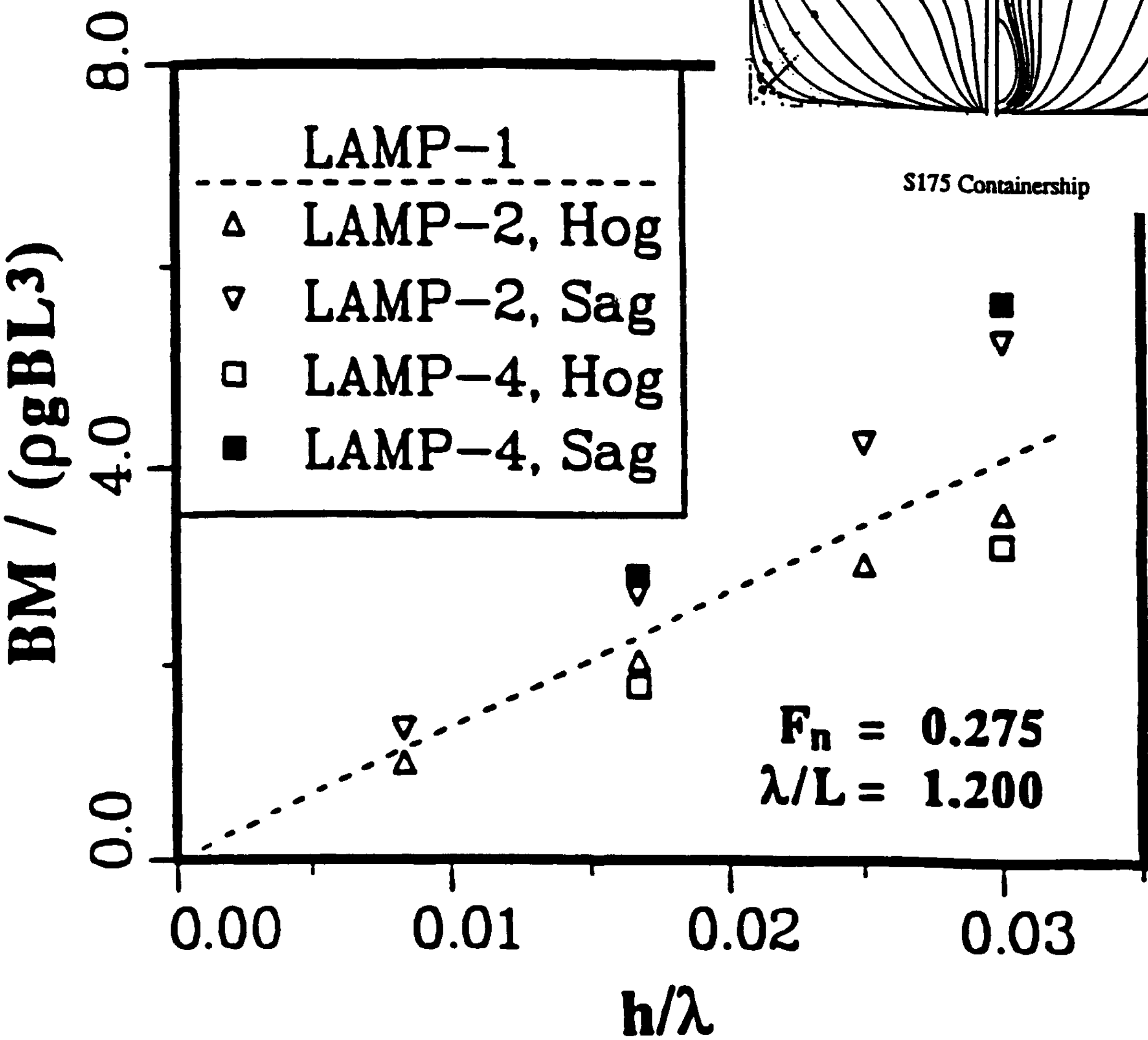
MIDSHIP BENDING MOMENT





Comparison Between Linear (LAMP-1), Approximate Nonlinear (LAMP-2) and Fully Nonlinear (LAMP-4)

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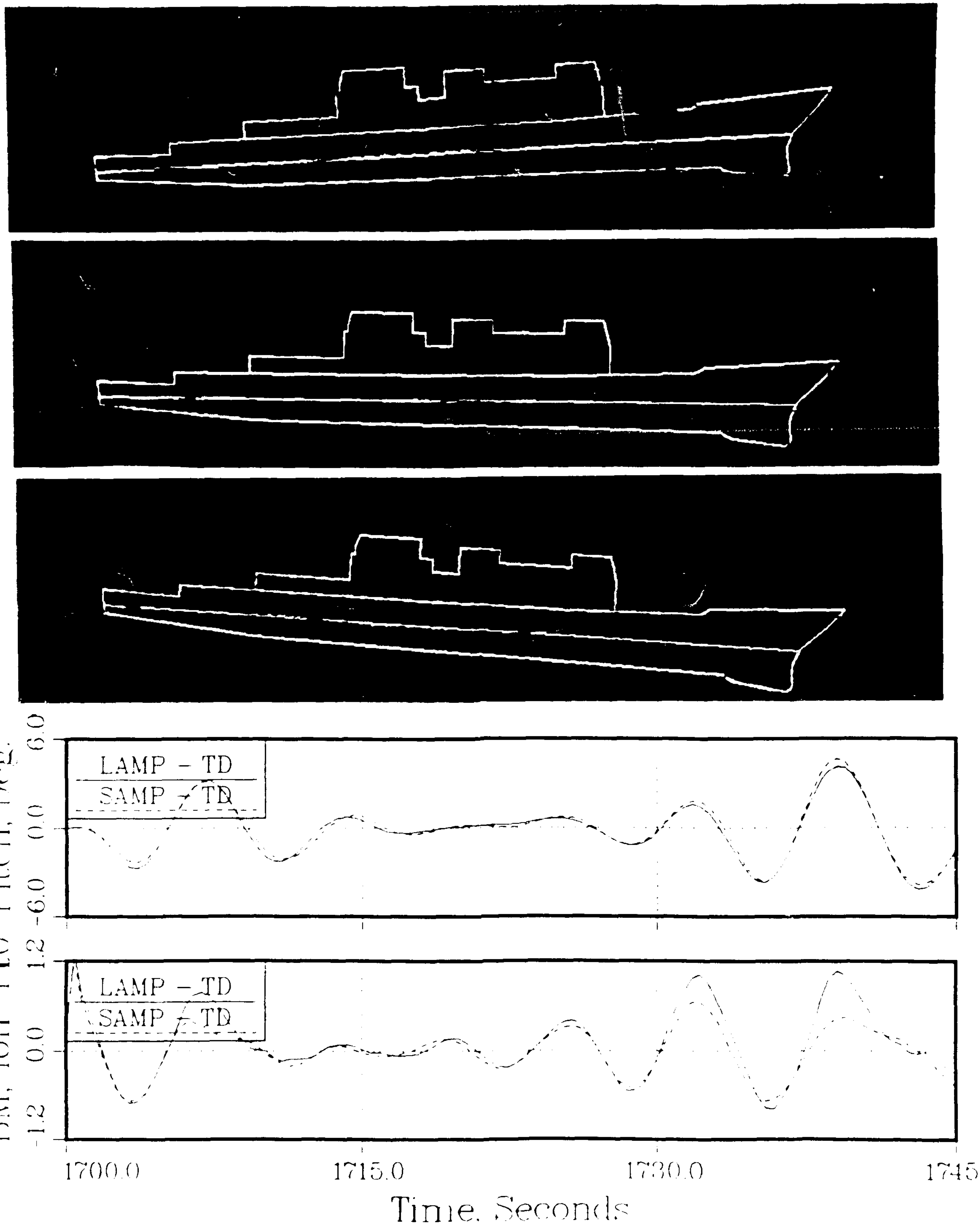
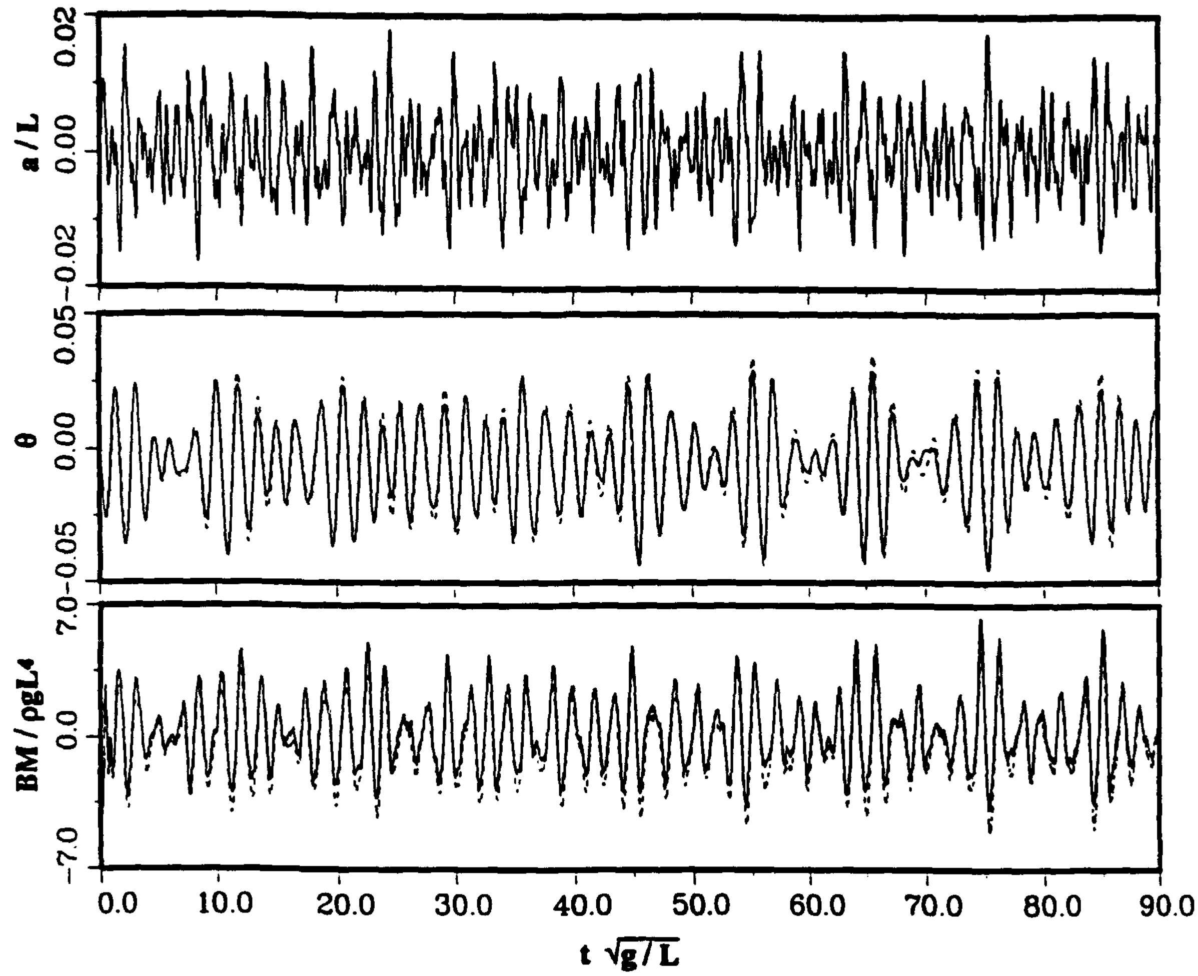
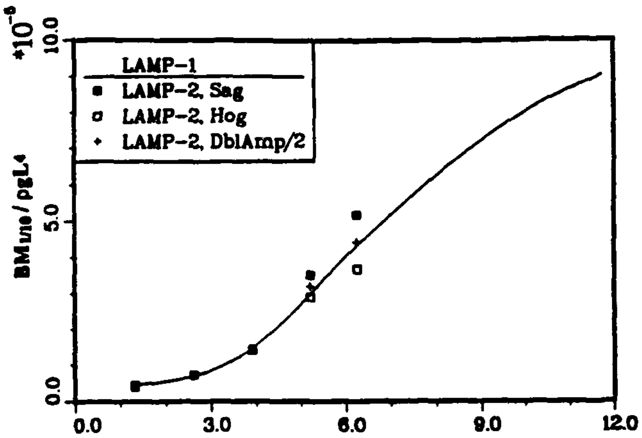
SAIC Large Amplitude Motion Program (LAMP)
AEGIS CRUISER IN WAVES

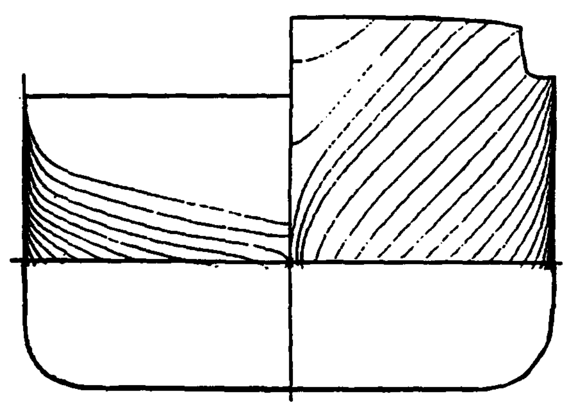




Time Record of Wave Elevation and Linear (LAMP-1,----) and Nonlinear (LAMP-2,—) Predictions of Pitch and Bending Moment for APL Containership at $F_n = 0.244$ in Unidirectional Irregular Head Seas with $h_{1/3} = 6.261$ meter.



$BM_{1/10}$ as a Function of $H_{1/3}$ for APL Containership at $F_n = 0.244$ in Unidirectional Irregular Head Seas



Body Plans for CG47 AEGIS Cruiser and APL Containership

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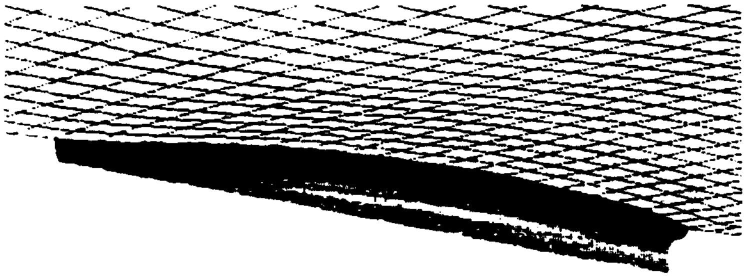
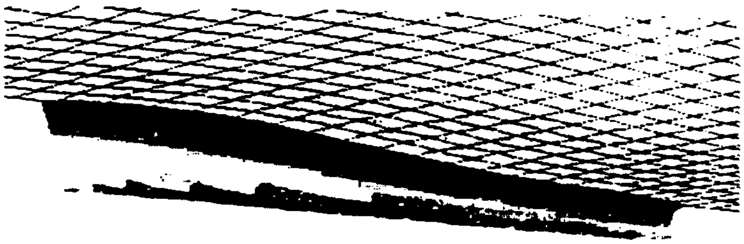
SIMULATION-BASED DESIGN

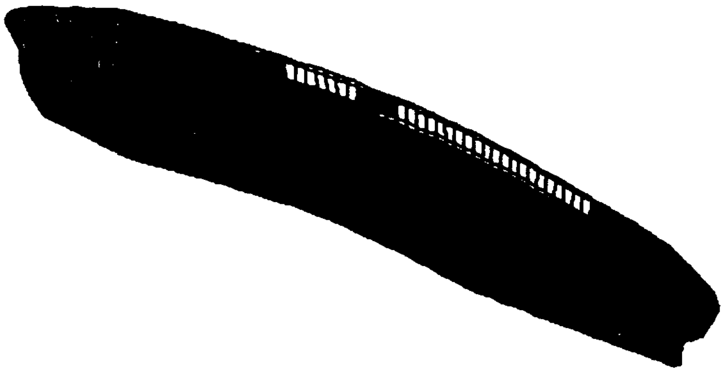
June '94 Demo
Sequence #7
Multi-Disciplinary Physics-Based Design

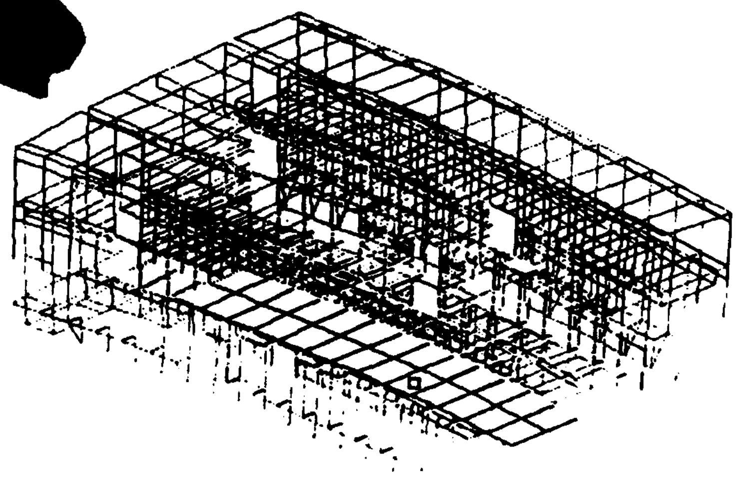
Hydrodynamic Wave Loads





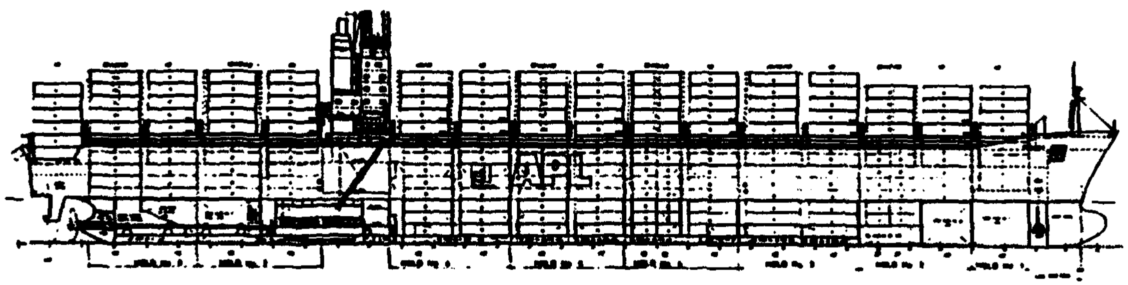
Structural Responses





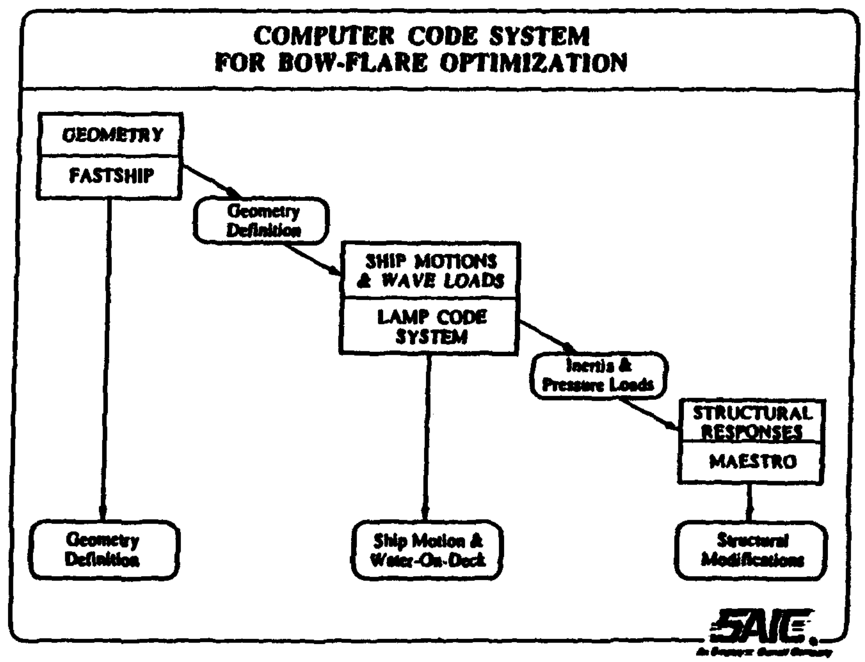
APPLICATION
ARPA HYPERCOMPUTING AND DESIGN (with Rutgers)

AMERICAN PRESIDENT LINE'S
"PRESIDENT TRUMAN" CONTAINERSHIP



SHIP HULL DESIGN OPTIMIZATION

SYNTHESIS MODEL
Life Cycle Profitability



NEEDED IMPROVEMENTS AND EXTENSIONS

Cooperative Project, NAVSEA and MARINTEK (1994-98)

1. Slamming Force and Global Structural Responses (Part I)
2. System Improvements and Integrations
3. System Efficiency
4. Validation for Typical Naval and Commercial Ships
5. Design Application Approach

Additional Important Improvements

6. Impact Load and Local Structural Responses (Part II)
7. Viscous Damping
8. Fully Nonlinear Effects

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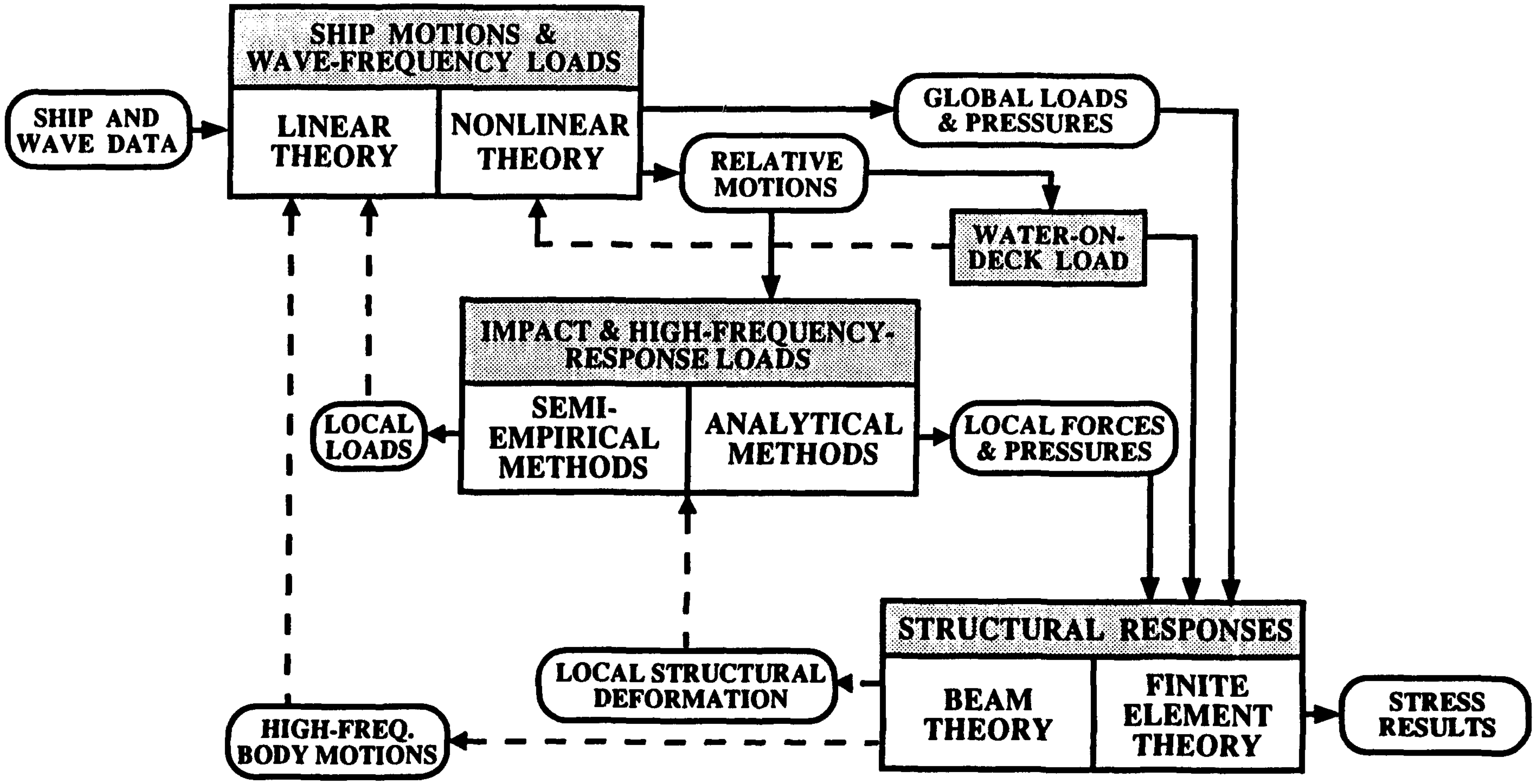
SIMULATION OF SHIP MOTIONS AND STRUCTURAL RESPONSES



MARINTEK SINTEF GROUP

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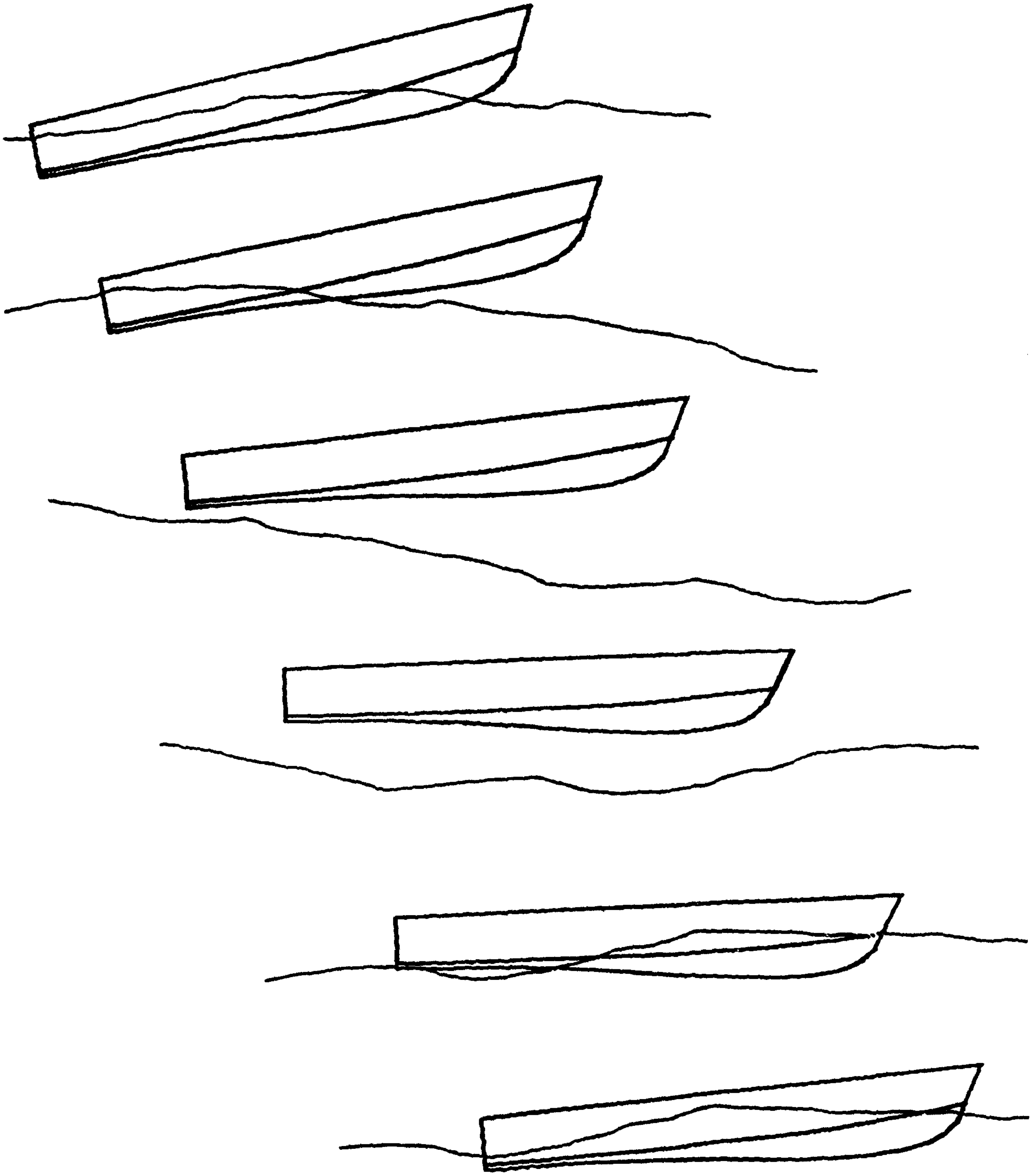
SIMULATION OF PLANING CRAFT MOTIONS

55' Patrol Boat at 40 knots in Sea State 5



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MARINTEK
SINTEF GROUP

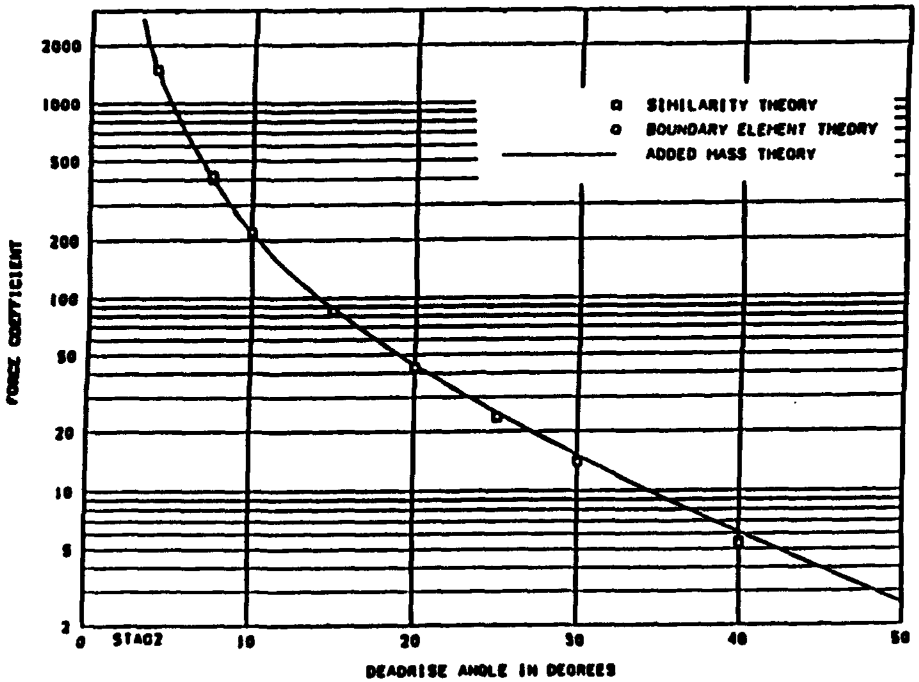
SIMPLAN - TIME-DOMAIN SIMULATION FOR PLANING CRAFT

SECTIONAL FORCE

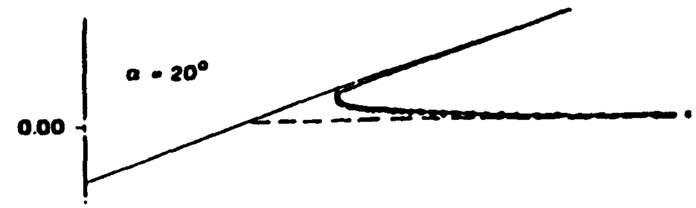
ADDED MASS THEORY

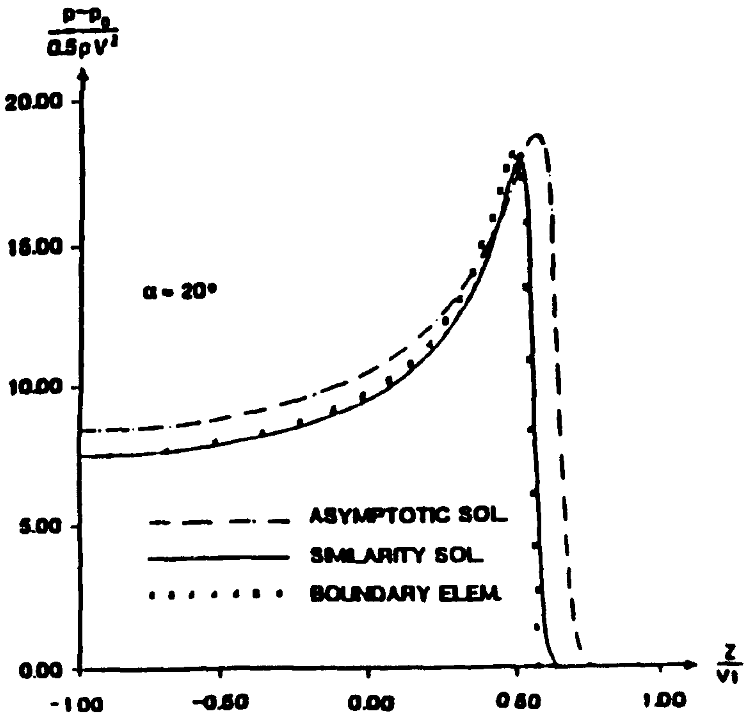
m′= 2-D sectional added mass

$$m' = C_m \frac{\pi}{2}\rho(\psi b)^2;\ C_m = 1 - \frac{\beta}{2\pi}$$



BOUNDARY ELEMENT THEORY





MARINTEK
SINTEF GROUP

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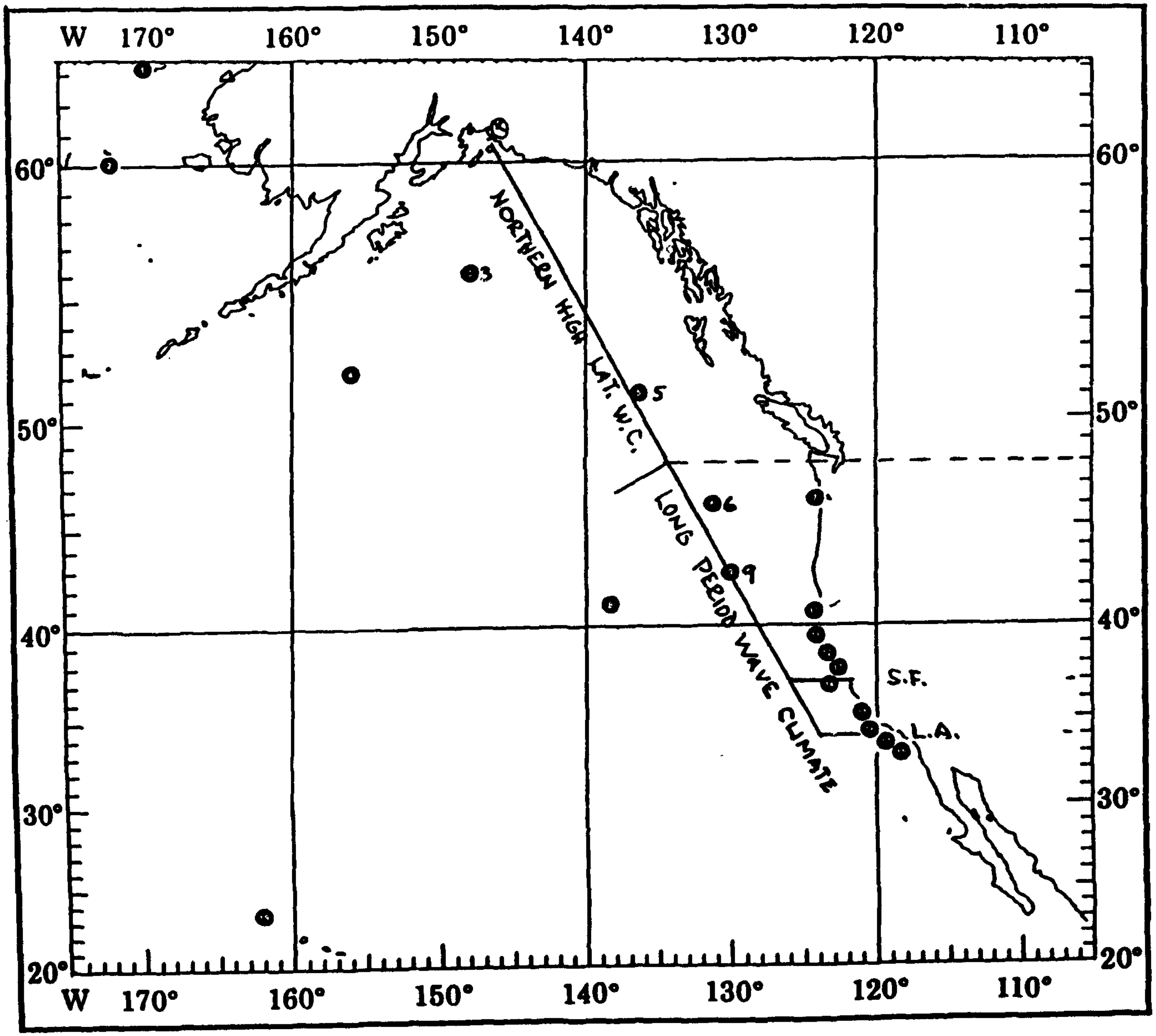
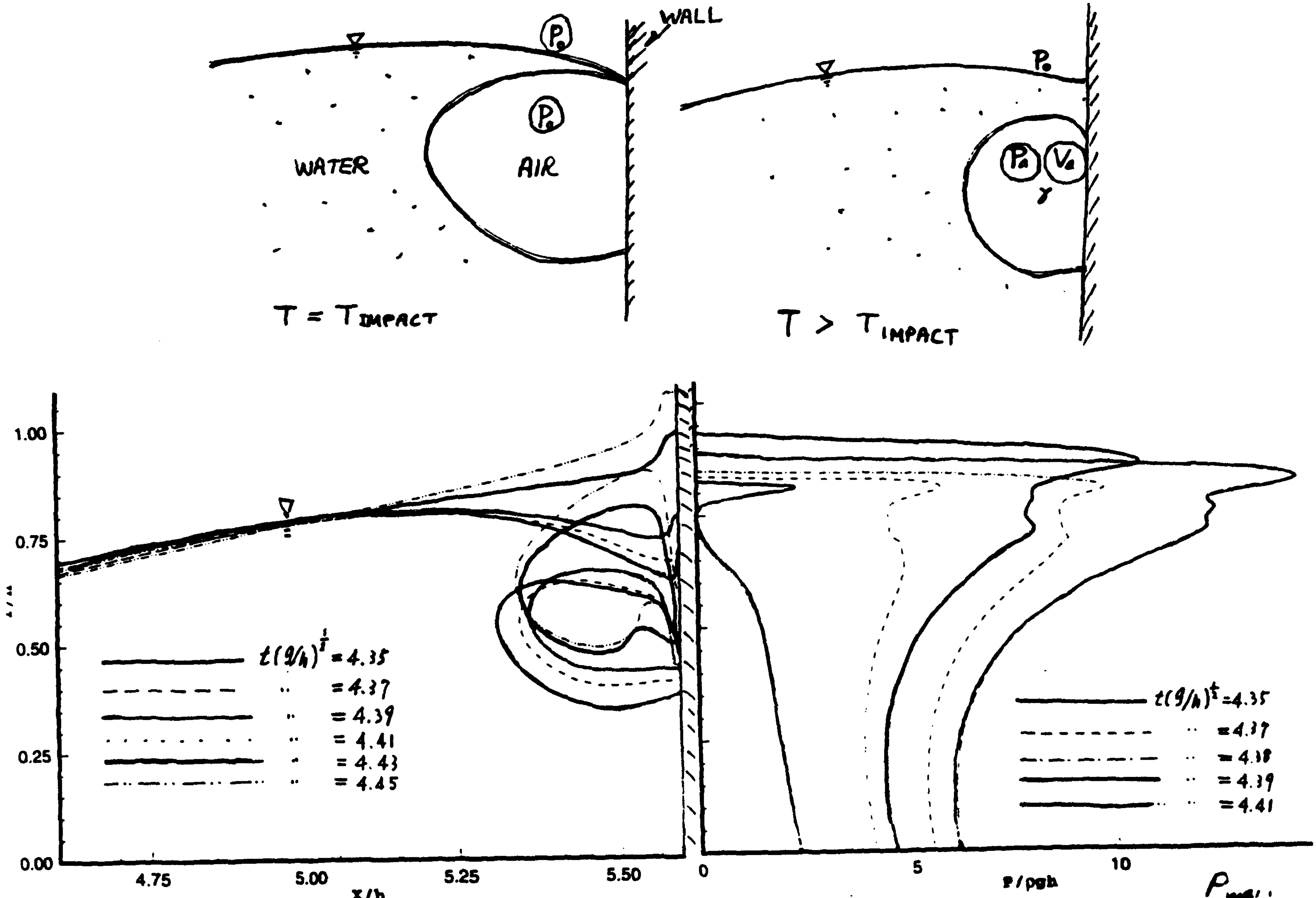
STRUCTURAL CRACKING PROBLEMS IN THE TRANS-ALASKA PIPELINE SERVICE (TAPS)



Plunging Wave Impact on a Vertical Wall



NONLINEAR VISCOUS ROLL DAMPING

Equivalent Linear Roll Damping (SMP)

$$\boxed{\text{Roll Viscous Damping} = [K\,|\dot{\eta}_4|\,] \cdot \dot{\eta}_4(t)}$$

K = damping coefficient

$|\dot{\eta}_4|$ = statistical average roll-velocity amplitude

Nonlinear Time-Domain Roll Damping (LAMP)

$$\boxed{\text{Roll Viscous Damping} = B^v_{44}(t) \cdot \dot{\eta}_4(t)}$$

$$B^v_{44}(t) = f(\eta_4(t), \dot{\eta}_4(t))$$

Table 1: Viscous and Lift Effects

Effect	Reference	Linearity
Hull Lift	Low Aspect Ratio Lifting Theory	Linear
Skeg, Bilge Keel and Foil Lift	High Aspect Ratio Lifting Theory	Linear
Hull Eddymaking	Tanaka (1960) and Ikeda et al. (1978)	Non-Linear
Bilge Keel Eddymaking	Kato (1966)	Non-Linear
Skeg and Foil Eddymaking	Hoerner (1958) and Ikeda et al. (1978)	Non-Linear
Hull Skin Friction	Kato (1958)	Non-Linear

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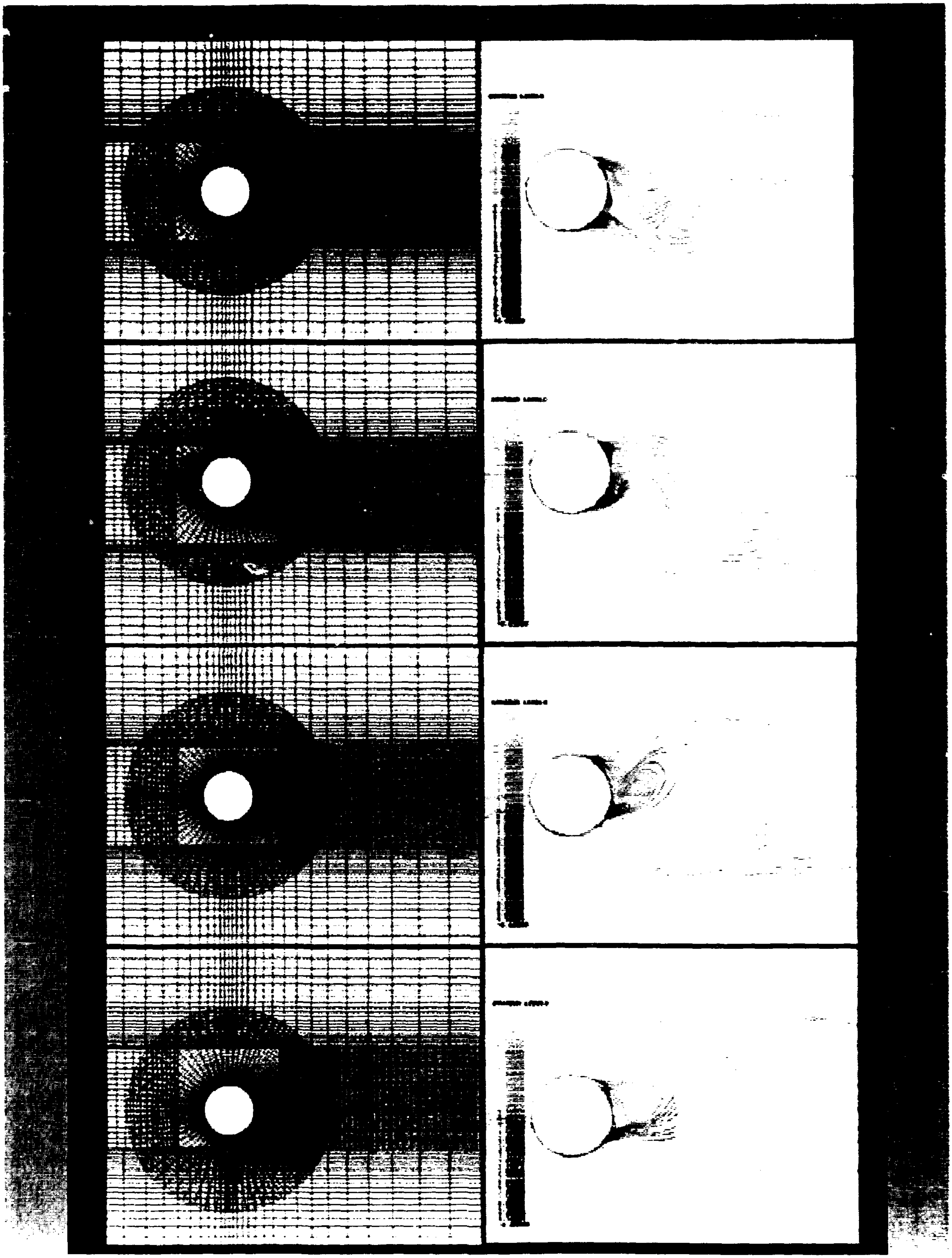
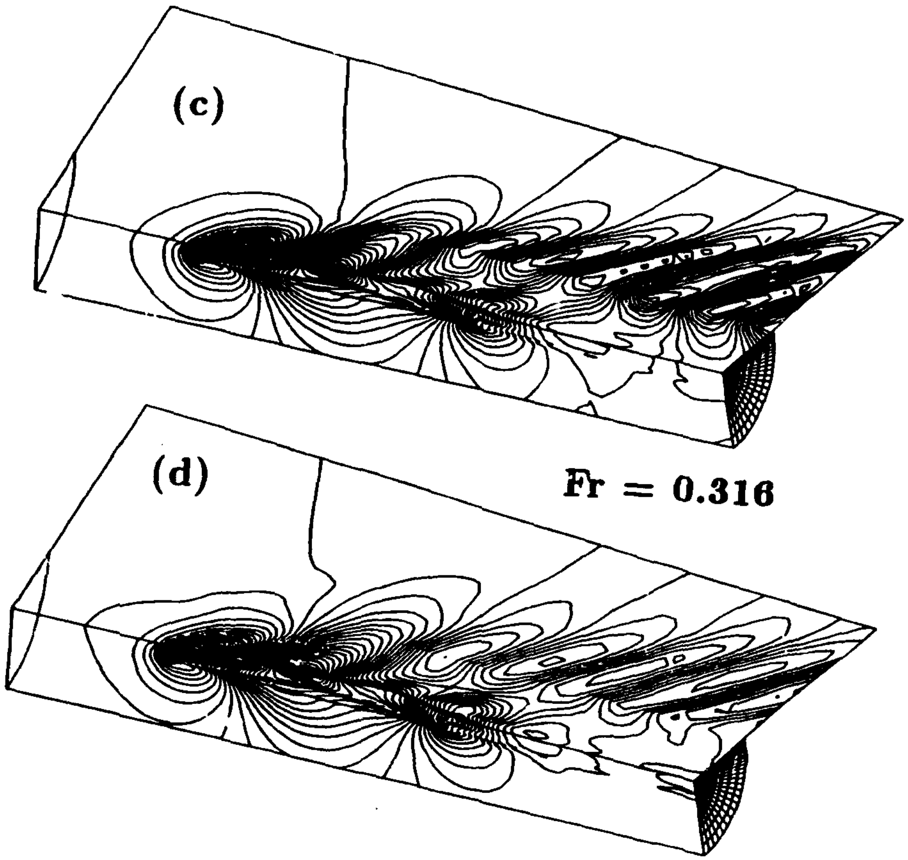
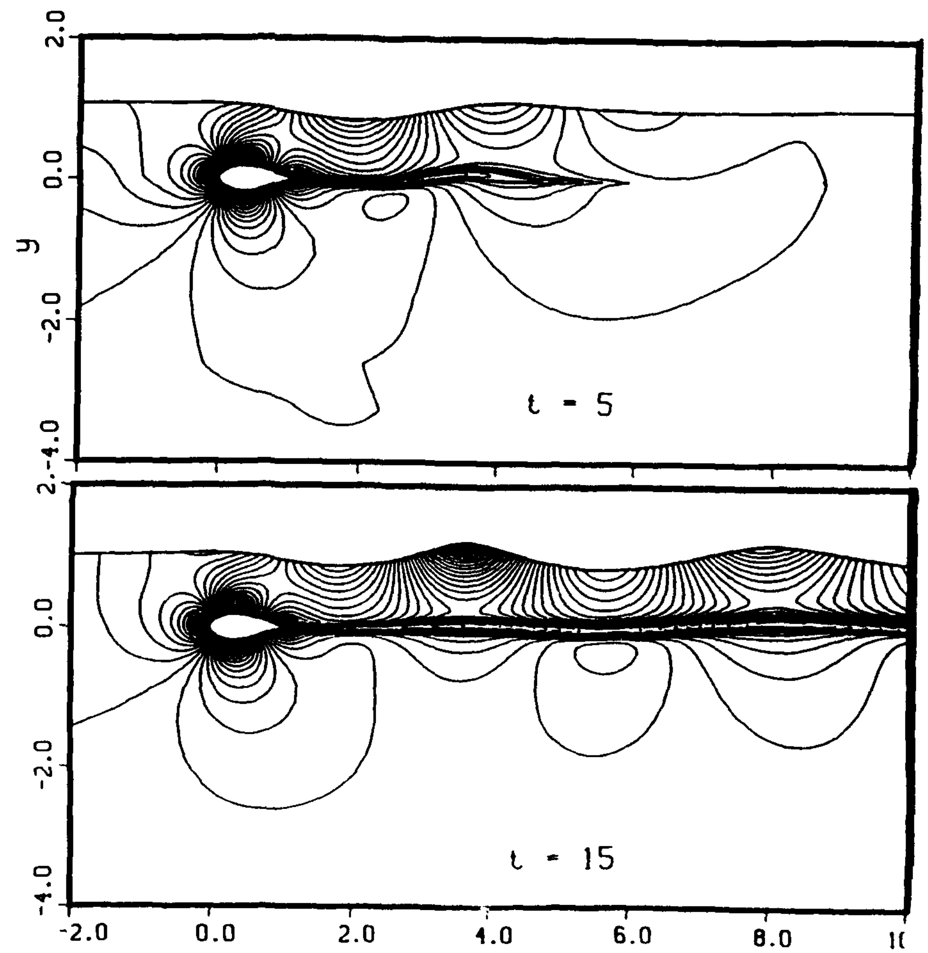


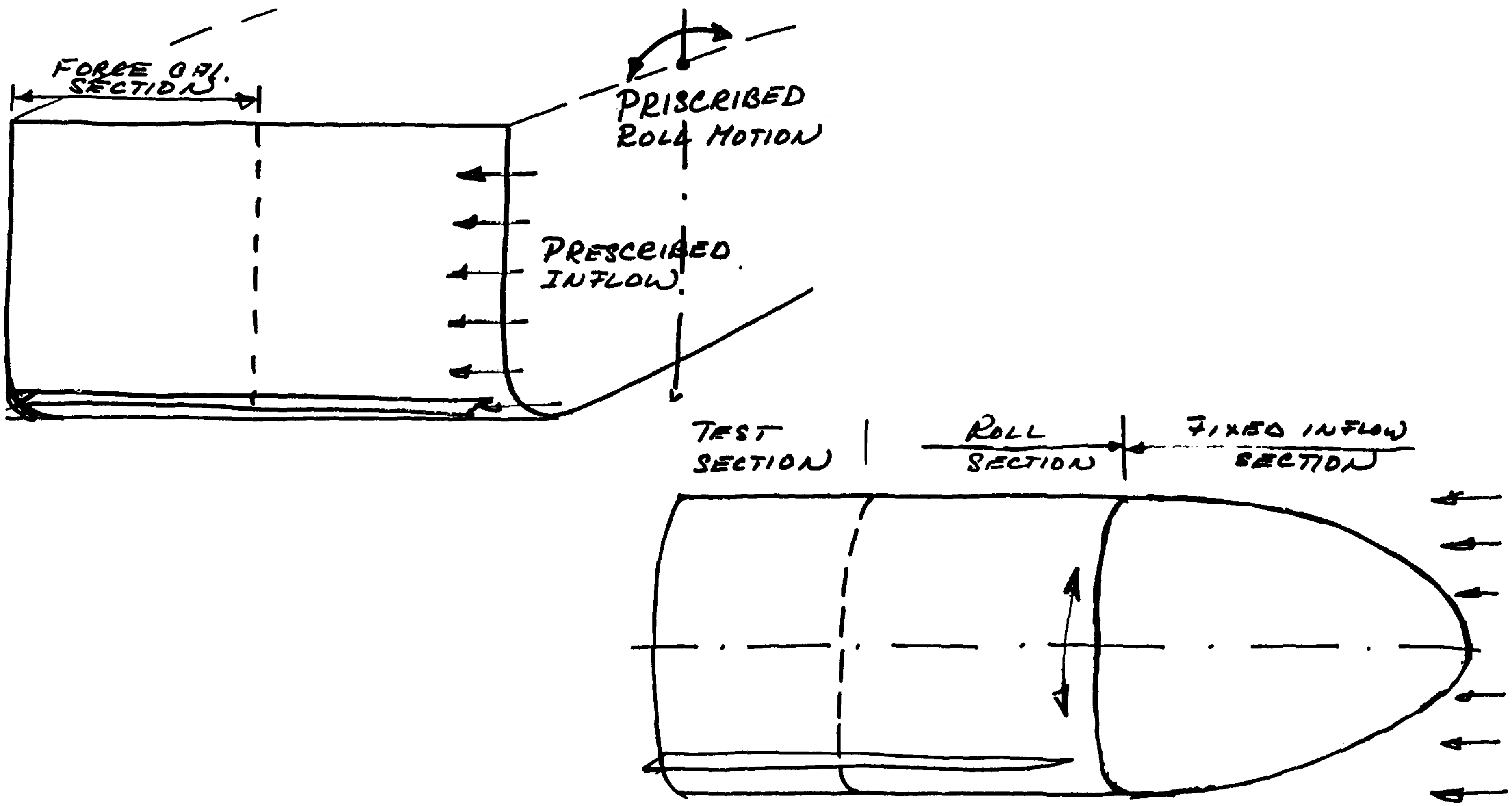
ONR NONLINEAR SEAKEEPING INITIATIVE
RANS/POTENTIAL FLOW COUPLING





ROLL DAMPING DUE TO 3-D COMBINED VISCOUS AND LIFT EFFECTS

APPROACH: **Calculate Bilge-Keel / Hull Damping by Unsteady 3-D RANS Code and "2-D Geometry"**



SHIP MOTIONS

AND

WAVE INDUCED STRUCTURAL LOADS

BY THE CODE

SWAN

by Paul D. Sclavounos
MIT Department of Ocean Engineering

Presentation to

ONR WORKSHOP

ON NONLINEAR SEA LOADS AND SHIP RESPONSE

A BASIS FOR SHIP STRUCRURAL DESIGN

College of Engineering
University of Michigan, Ann Arbor

July 7&8, 1994

REVIEW

OF SHIP MOTION

AND STRUCTURAL LOAD

PREDICTIONS

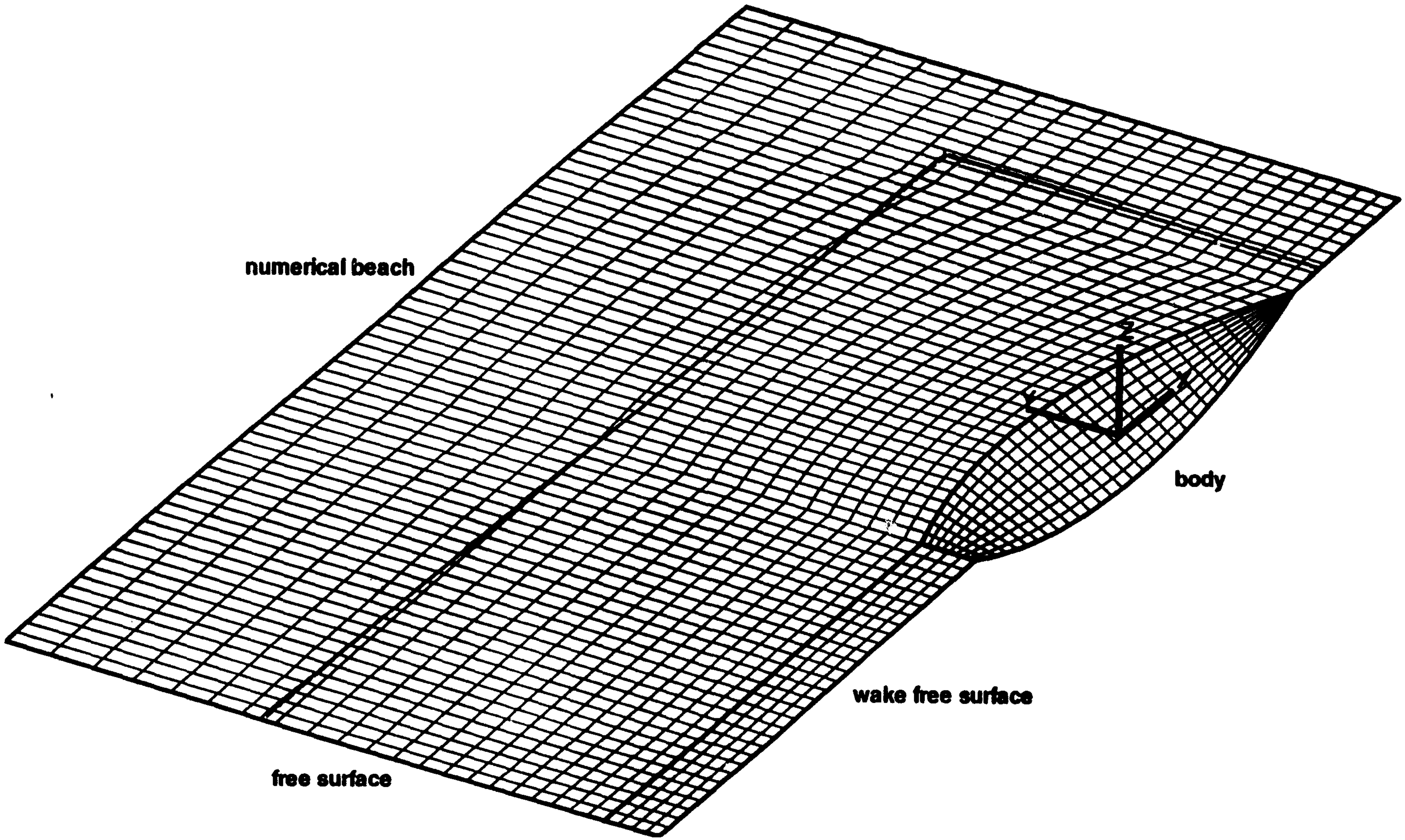
BY THE

FREQUENCY DOMAIN CODE

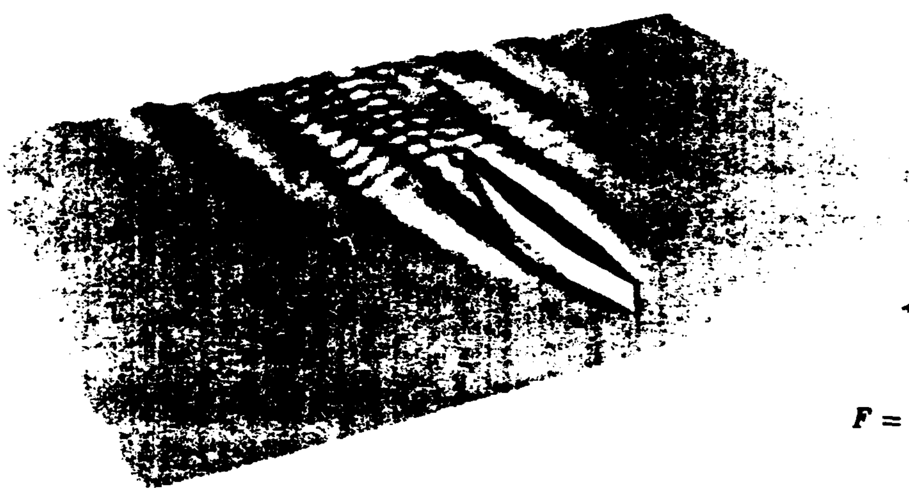
SWAN-1

SWAN2

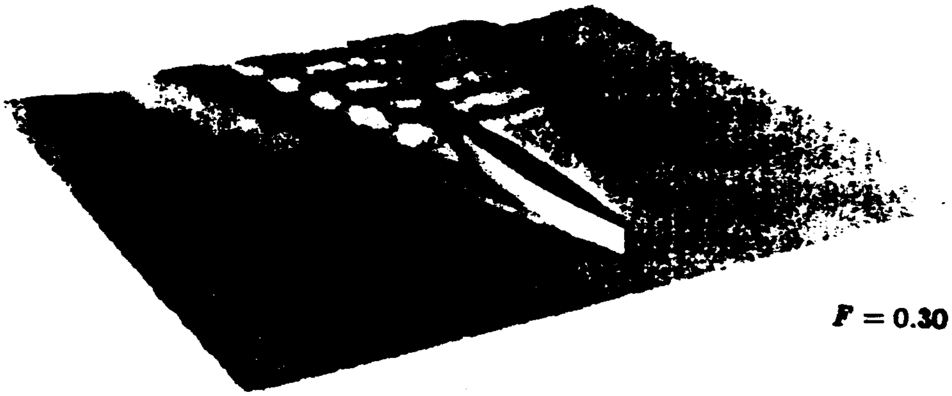
transom hull mesh



$F = 0.20$, $\omega\sqrt{\frac{L}{g}} =$ [illegible].0



$F = 0.20$, $\omega\sqrt{\frac{L}{g}} = 5.0$



$F = 0.30 \quad , \quad \omega\sqrt{\frac{L}{g}} = 3.0$

$Fr = 0.3 \quad , \quad \omega\sqrt{\frac{L}{g}} = 5.0$

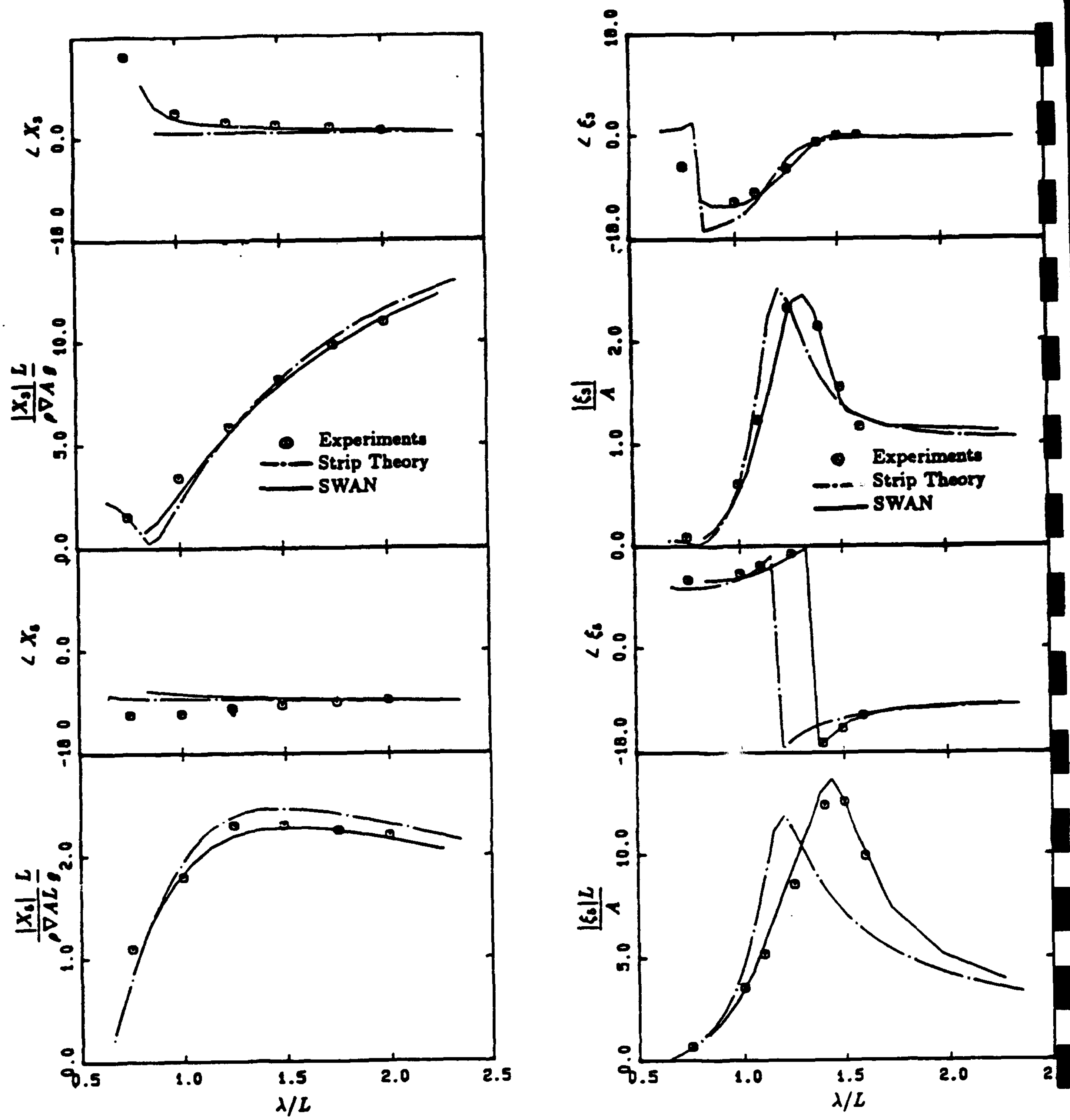


Figure 10 : Heave and pitch exciting forces and motions of a modified Wigley model advancing at Froude number $F = 0.3$ through regular head waves.

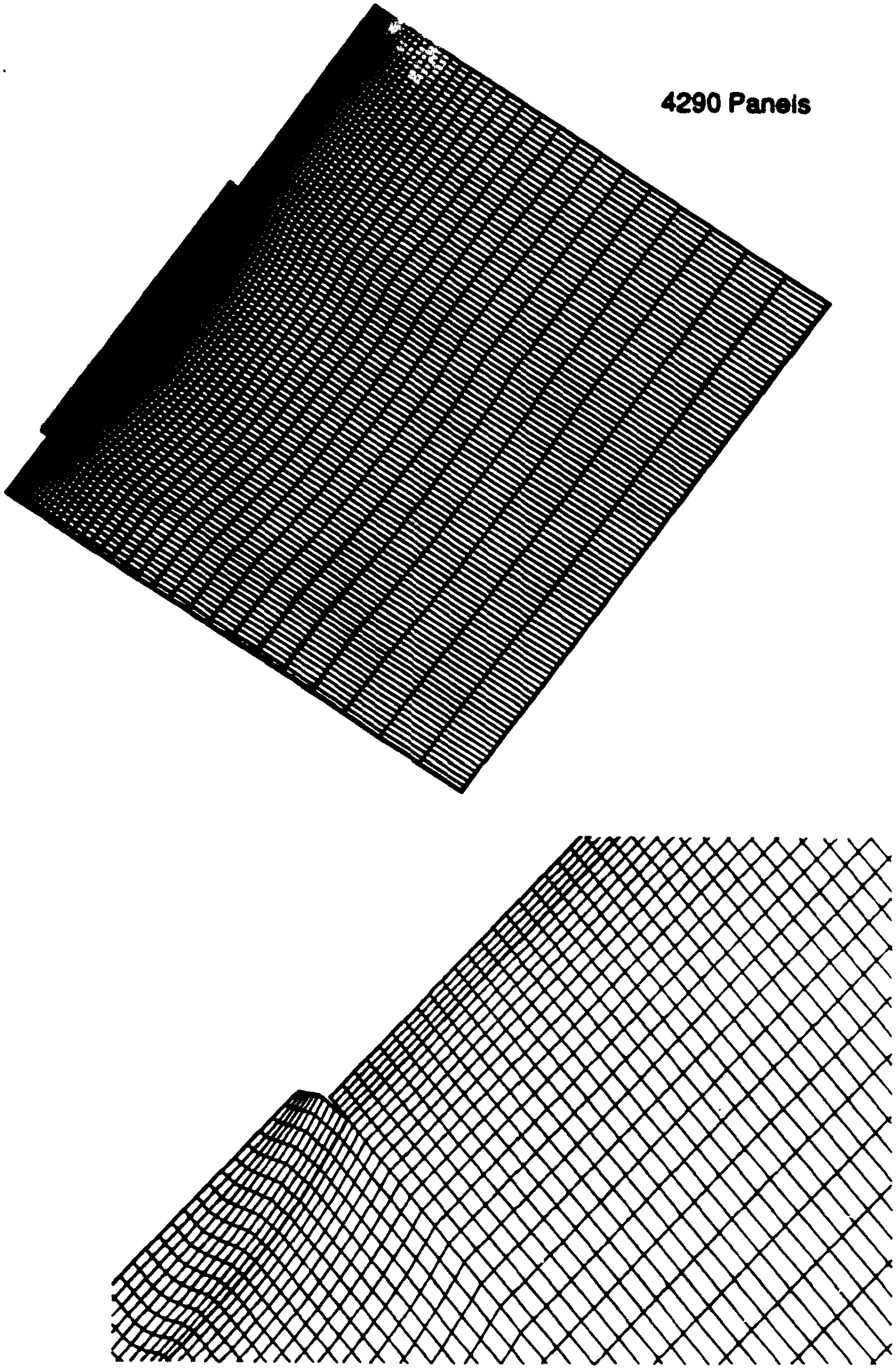
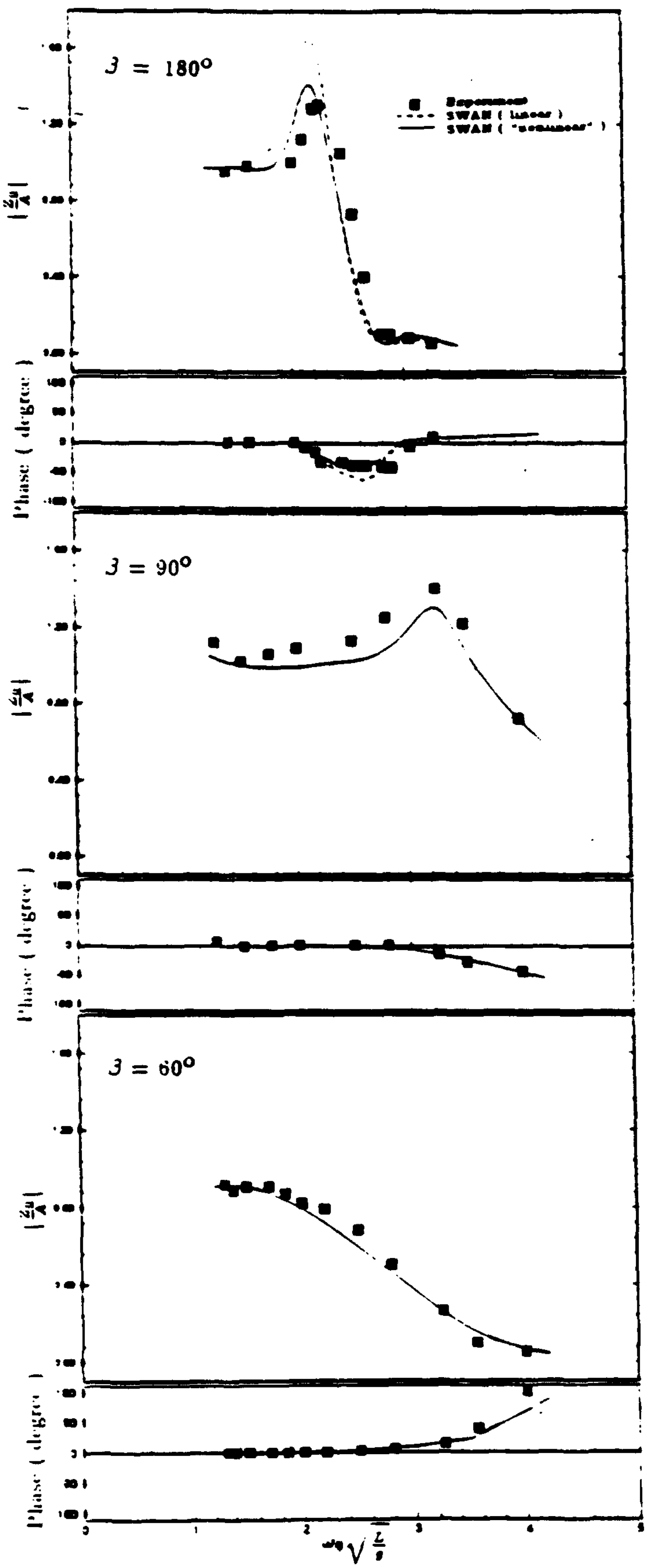


Figure 2 : Hull and Free Surface Discretization for S-175 Hull.



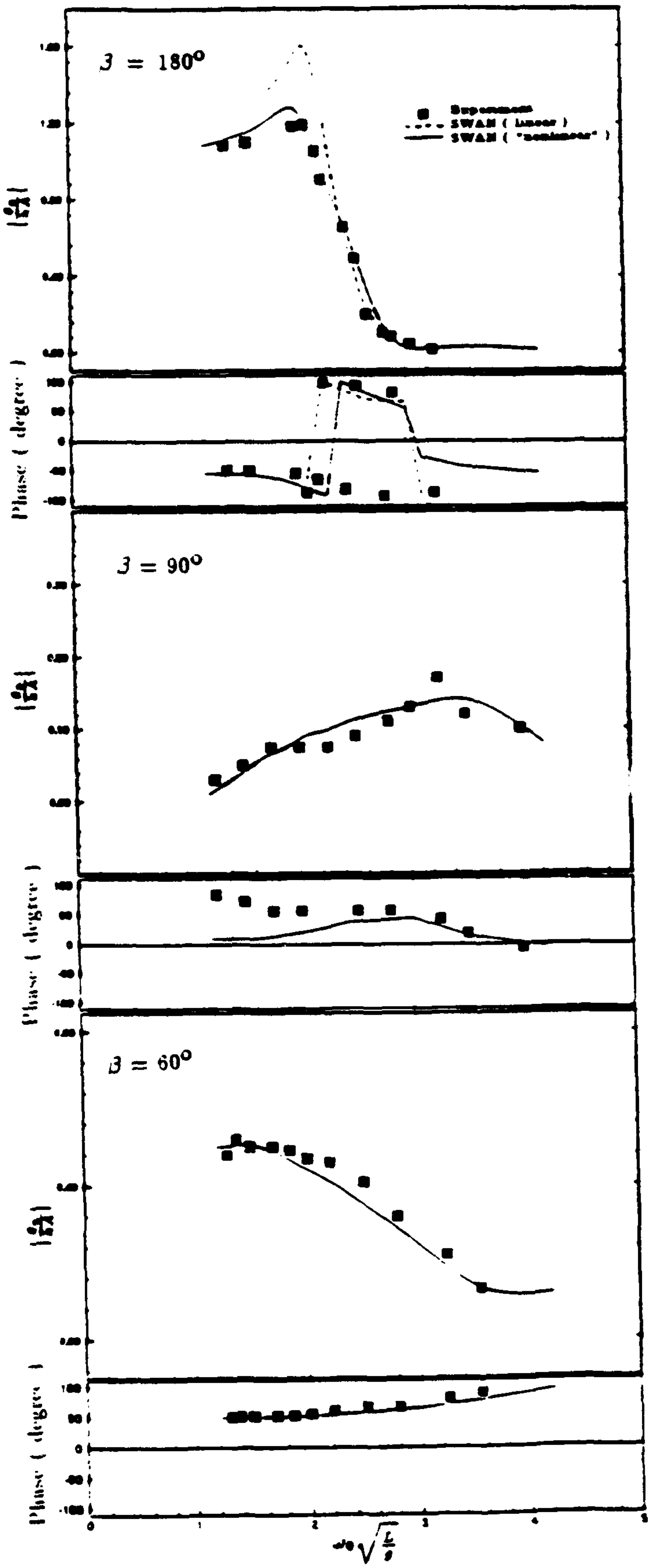
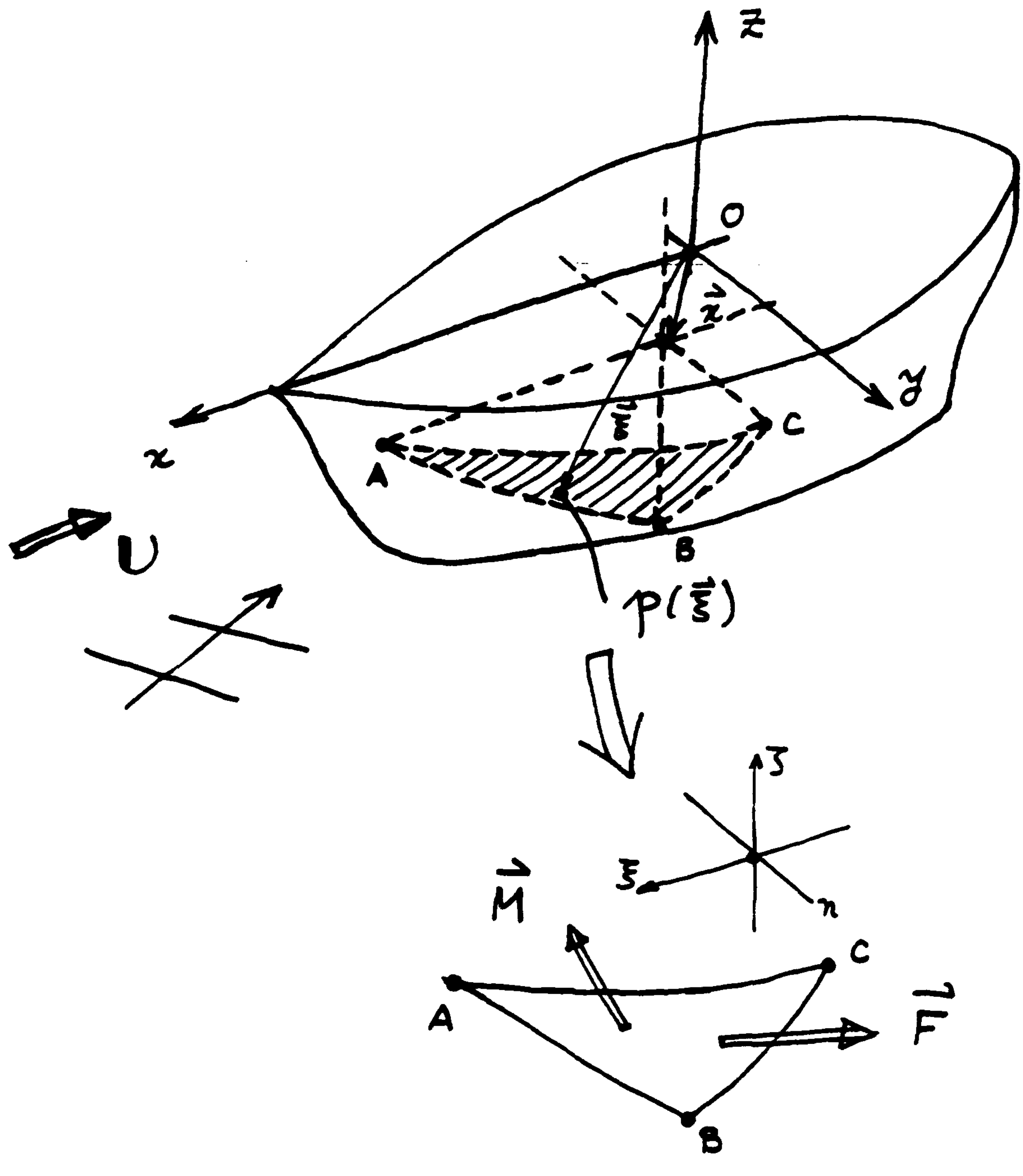


Figure 10 : Heave and Pitch Motions for the S-175 Hull in Head, Beam and Quartering Waves at Fr = 0.275.

STRUCTURAL LOADS EVALUATED BY

SWAN



$\vec{F} = (F_\xi, F_\eta, F_\zeta) \equiv$ SHEAR FORCE

$\vec{M} = (M_\xi, M_\eta, M_\zeta) \equiv$ BENDING MOMENT

ABOUT (ξ, η, ζ) FRAME

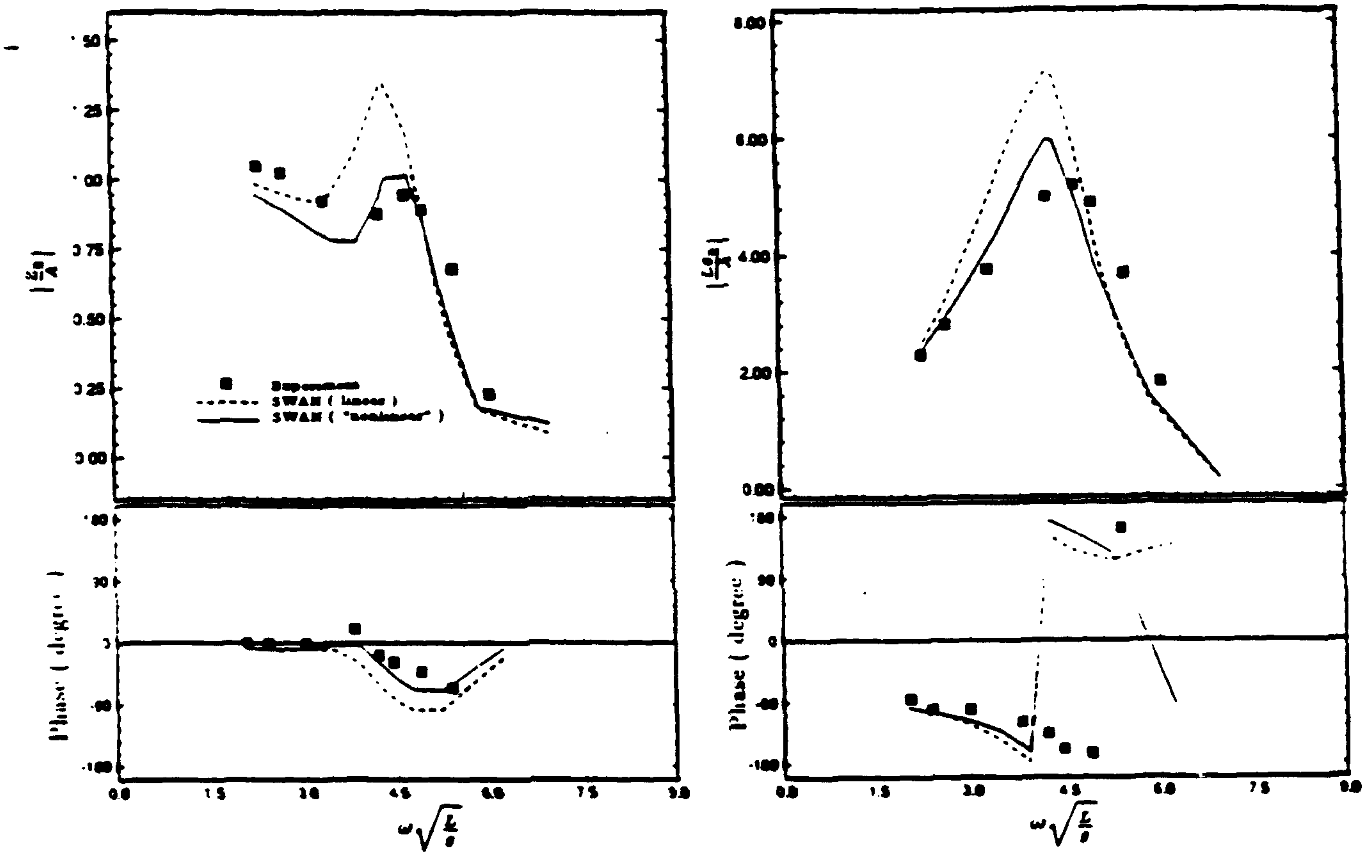


Figure 8 : **Heave and Pitch Motions for the SL-7 Hull in Head Waves at Fr = 0.3.**

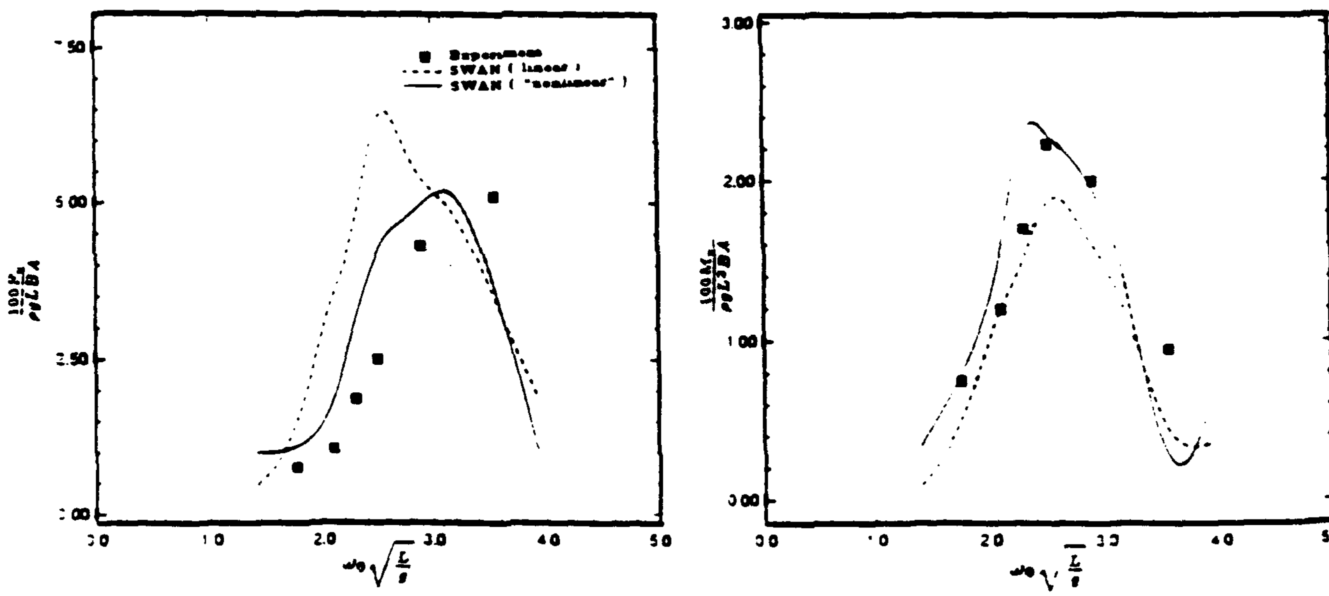


Figure 9 : **Vertical Shear Force and Bending Moment at Midship Section of the SL-7 Hull in Head Waves at Fr = 0.3.**

SEAKEEPING BY THE

TIME DOMAIN CODE

SWAN-2

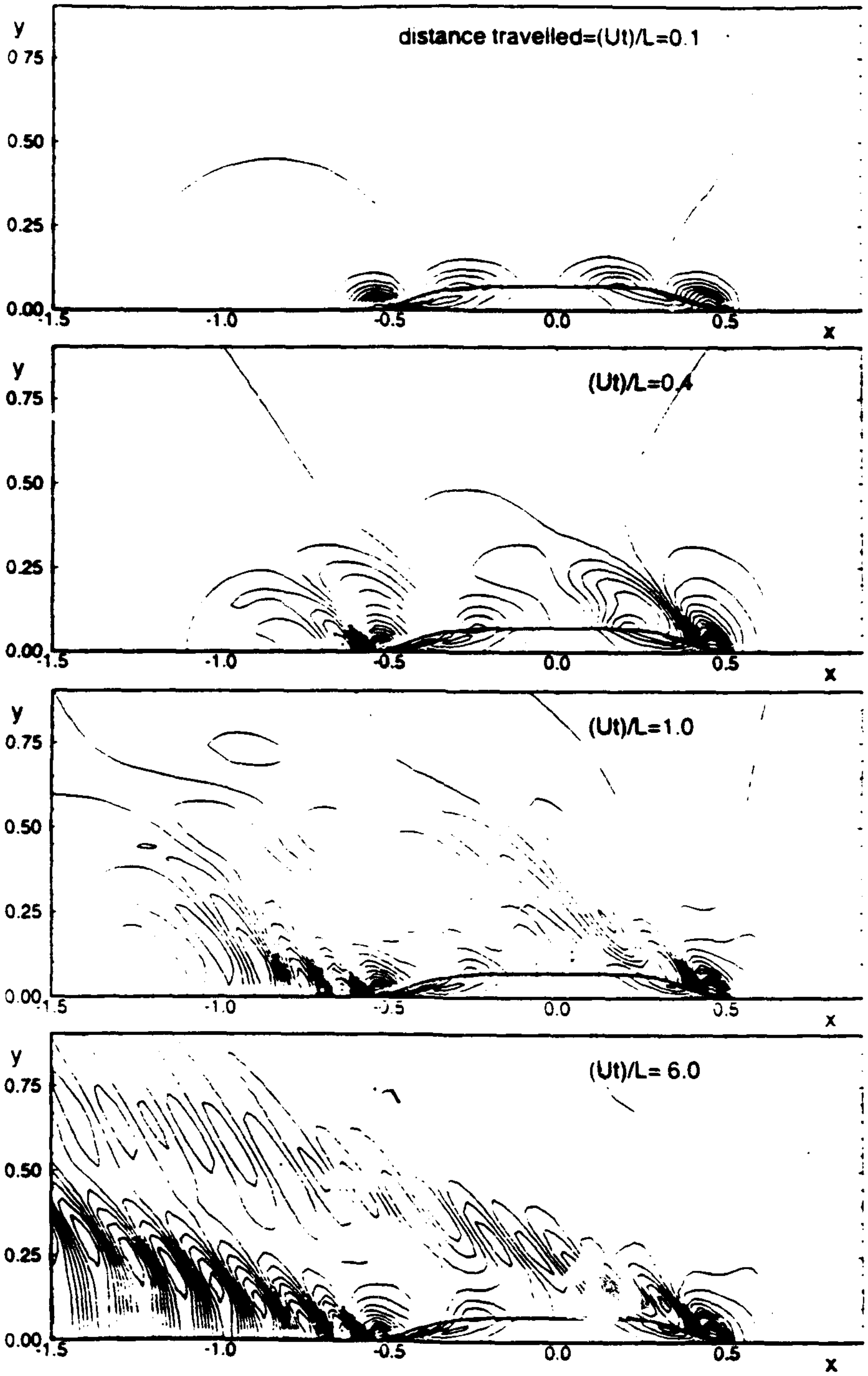
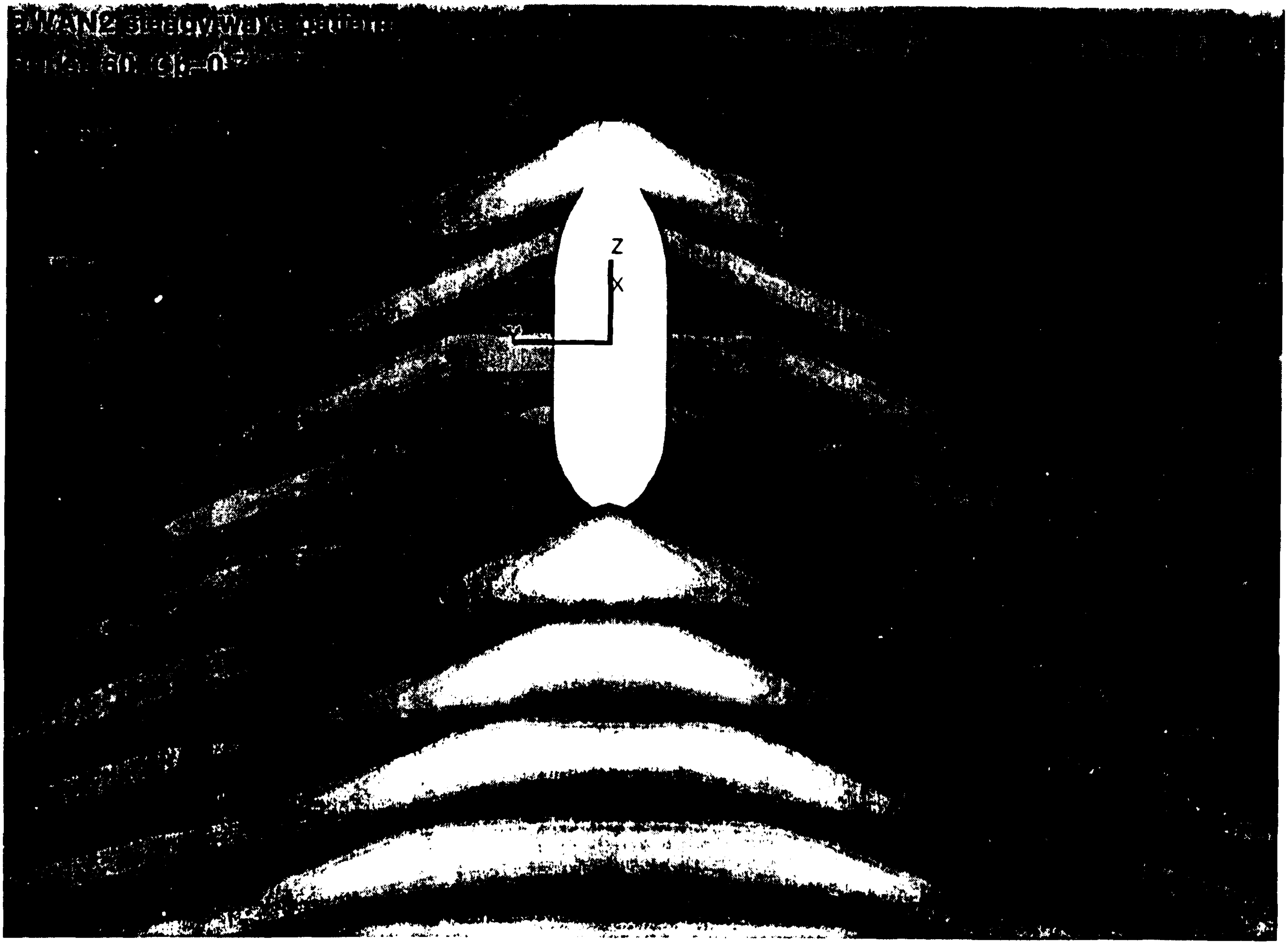
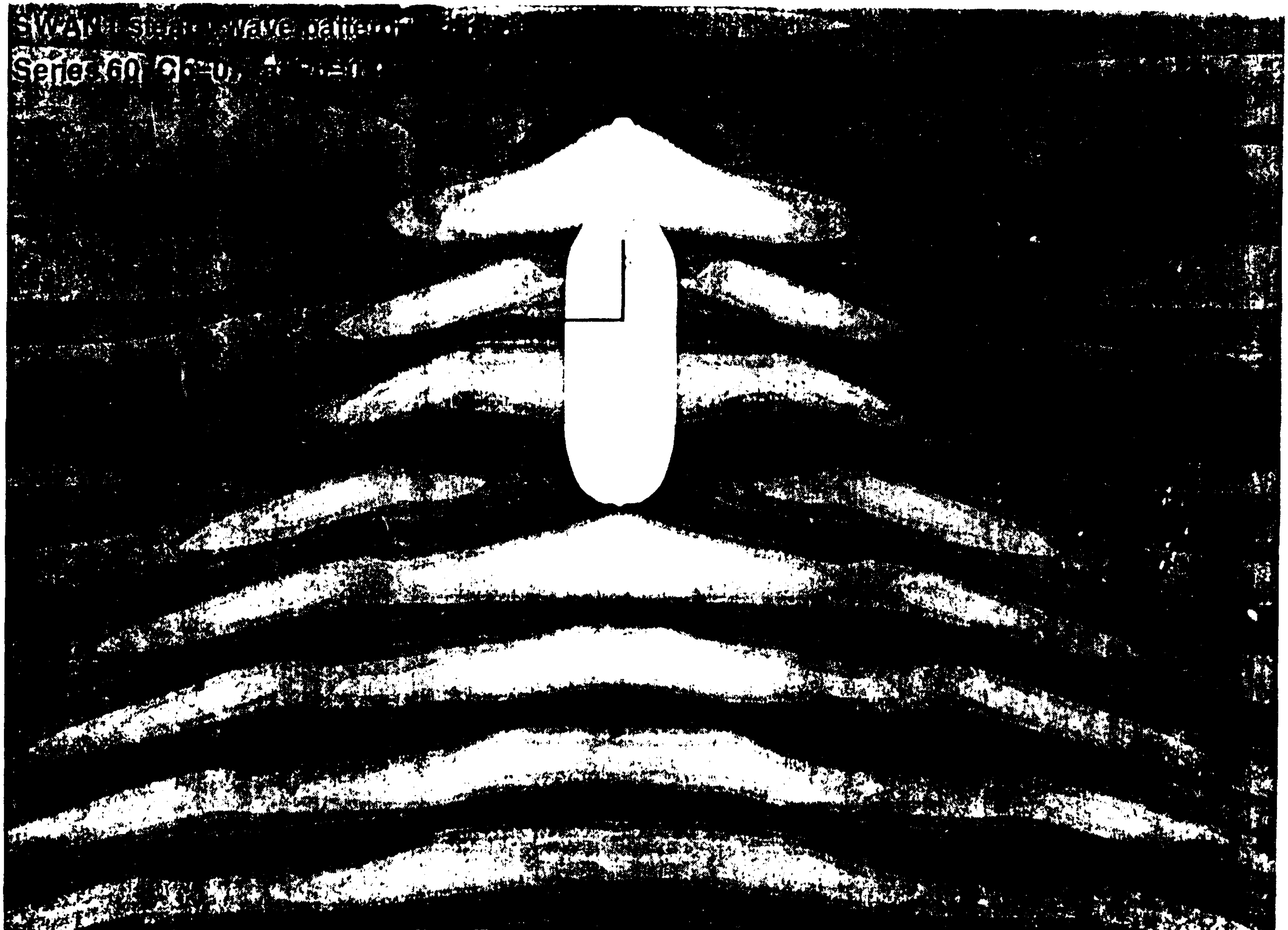


Figure 5-2: Transient wave elevation and body pressure contours for the Series 60 hull in steady forward motion at $\mathcal{F} = 0.2$, started impulsively from rest at $t = 0$.


SWAN2 steady wave pattern
Z
X


Series 60

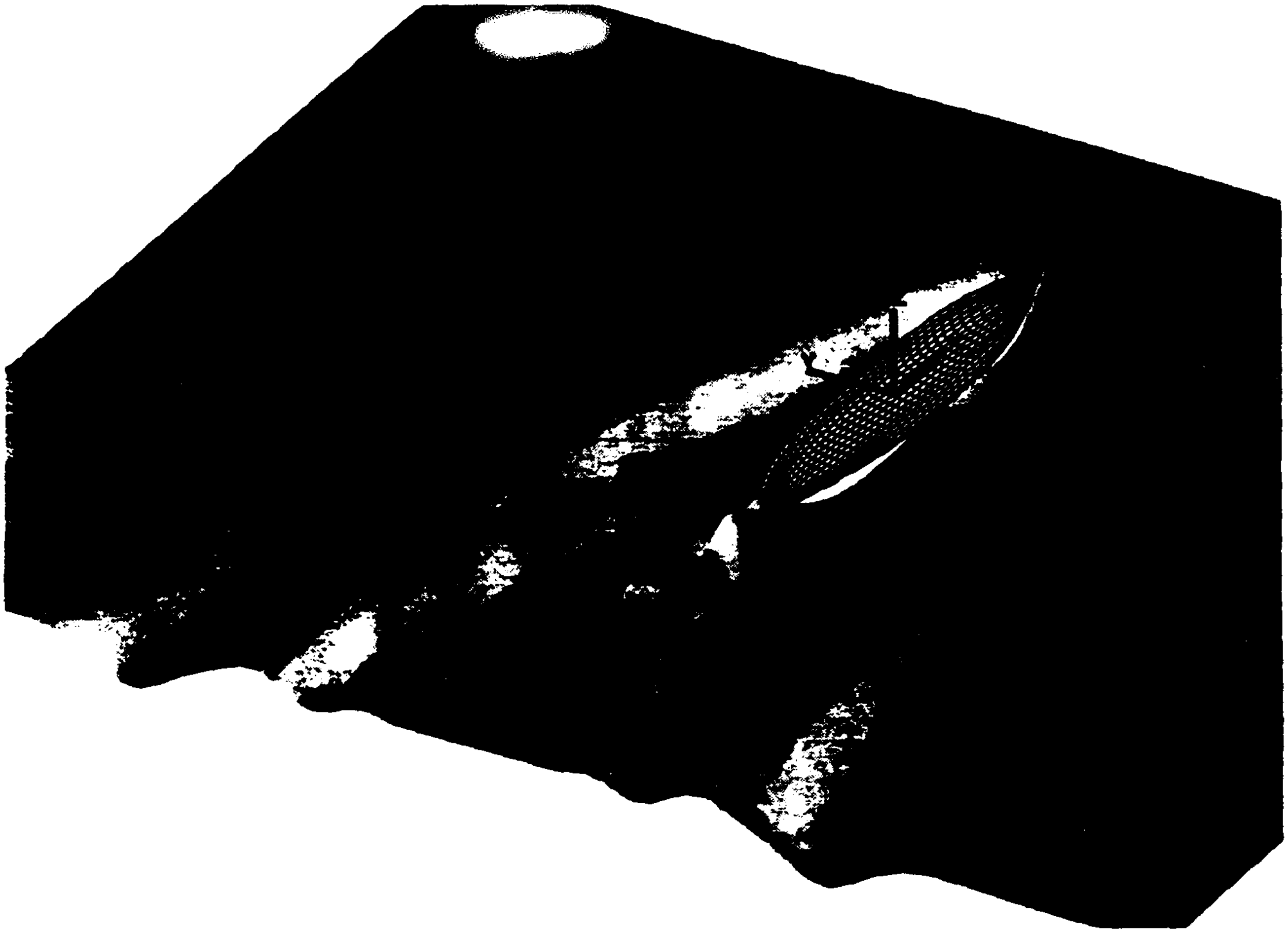


Figure 5-8: Radiated wave pattern for the Series 60 hull in forced. periodic heave at $\mathcal{F} = 0.2$ and encounter frequency $\omega/(g/L)^{\frac{1}{2}} = 3.335$, viewed from above and behind the vessel. A snapshot of the steady-state periodic wave pattern taken at the middle of the heave cycle.

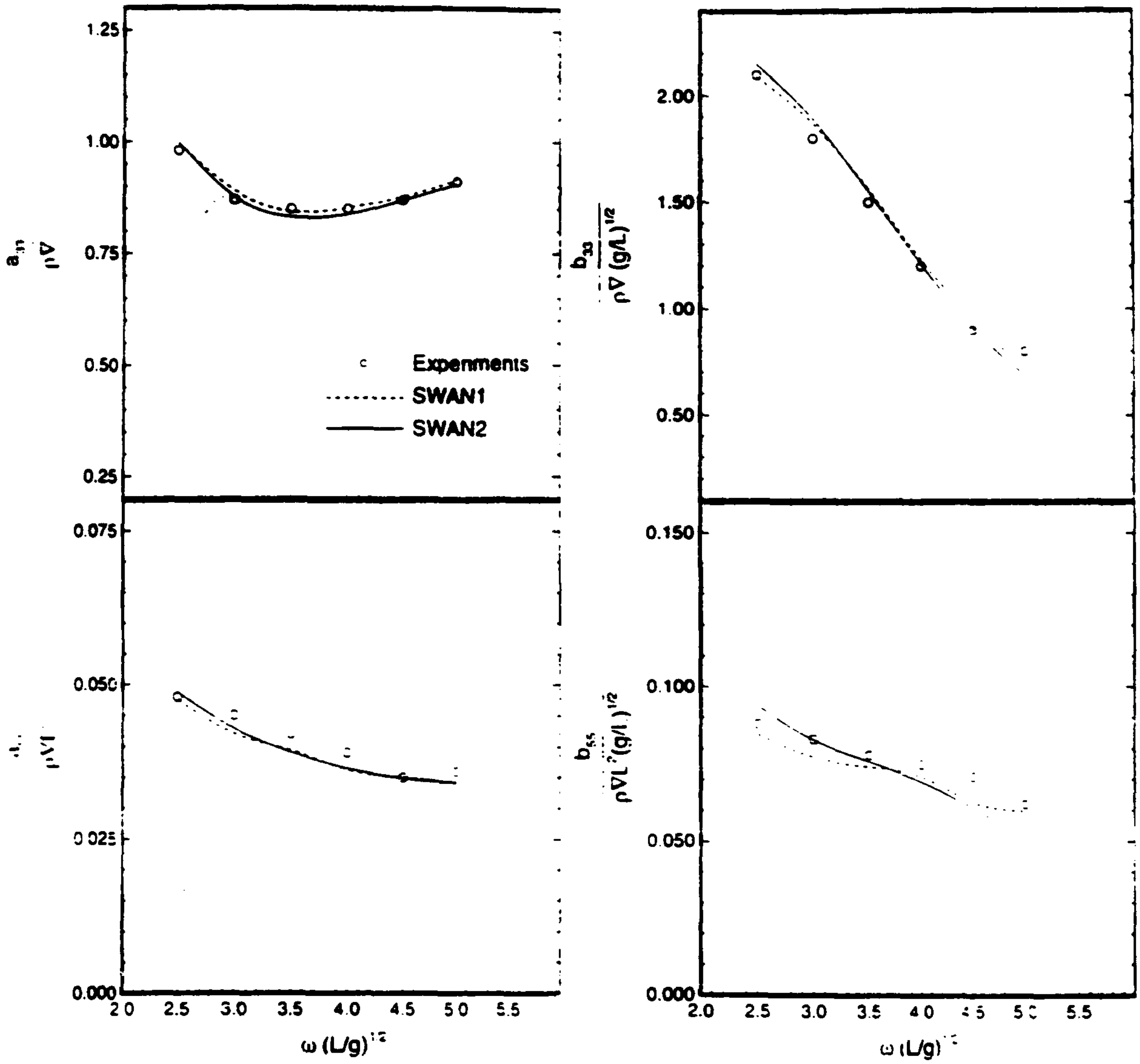


Figure 5-12: Diagonal added mass and damping coefficients for the Series 60 hull in heave and pitch at $\mathcal{F} = 0.2$.

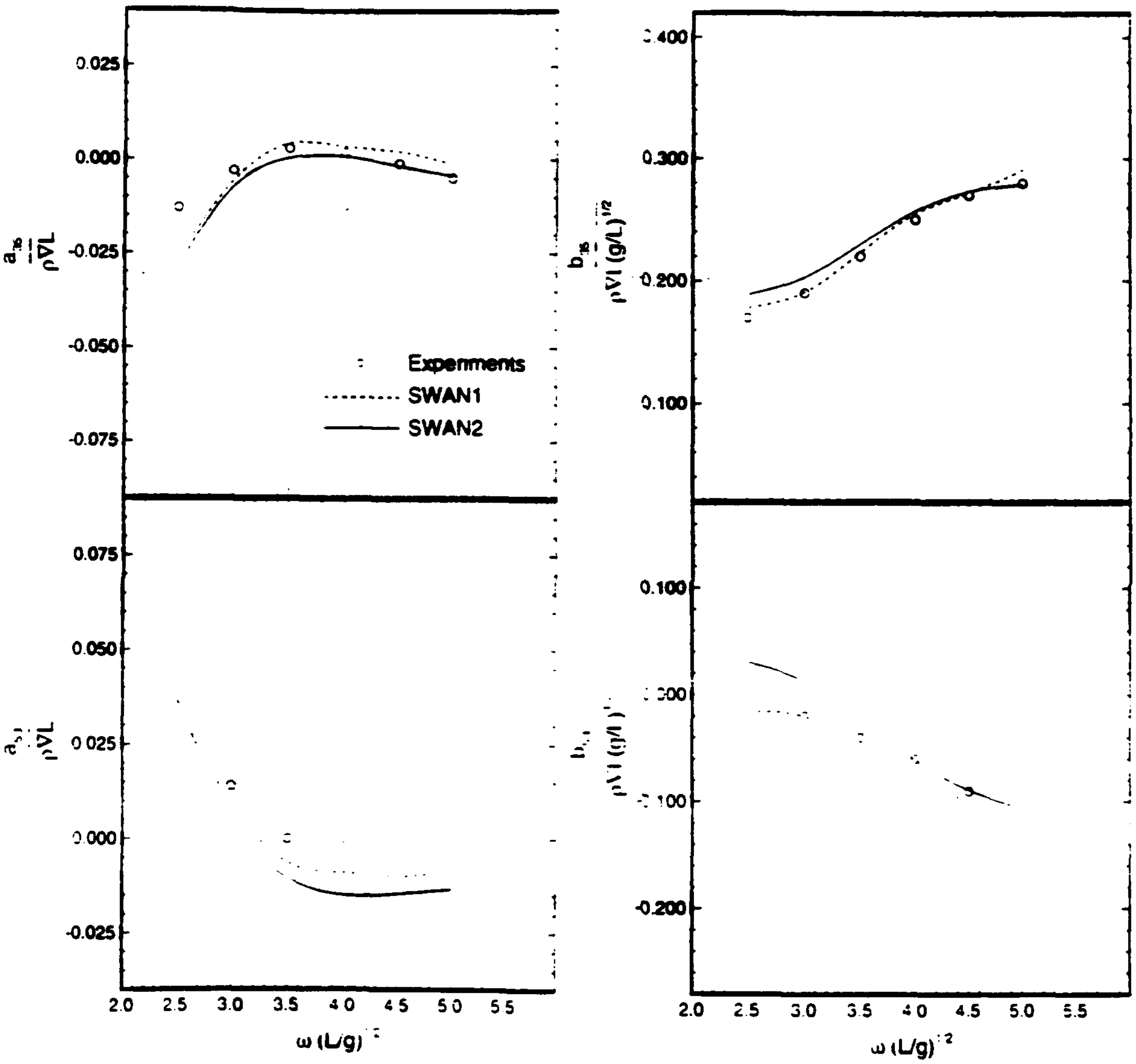


Figure 5-13: Cross-coupling added mass and damping coefficients for the Series 60 hull in heave and pitch at $\mathcal{F} = 0.2$.

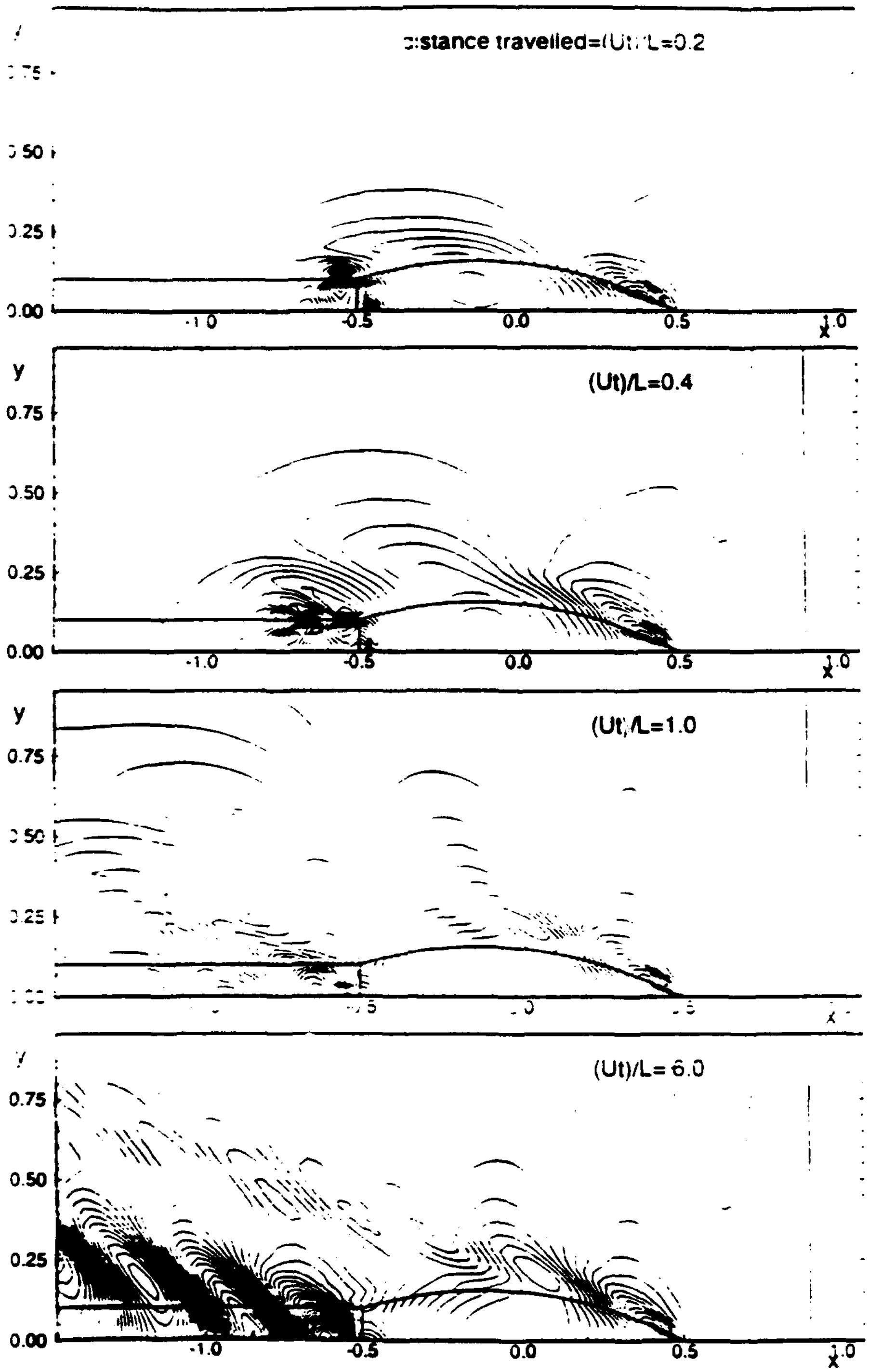


Figure 5-15: Transient wave elevation and body pressure contours for a transom hull in steady forward motion at $\mathcal{F} = 0.3$, started impulsively from rest at $t = 0$.

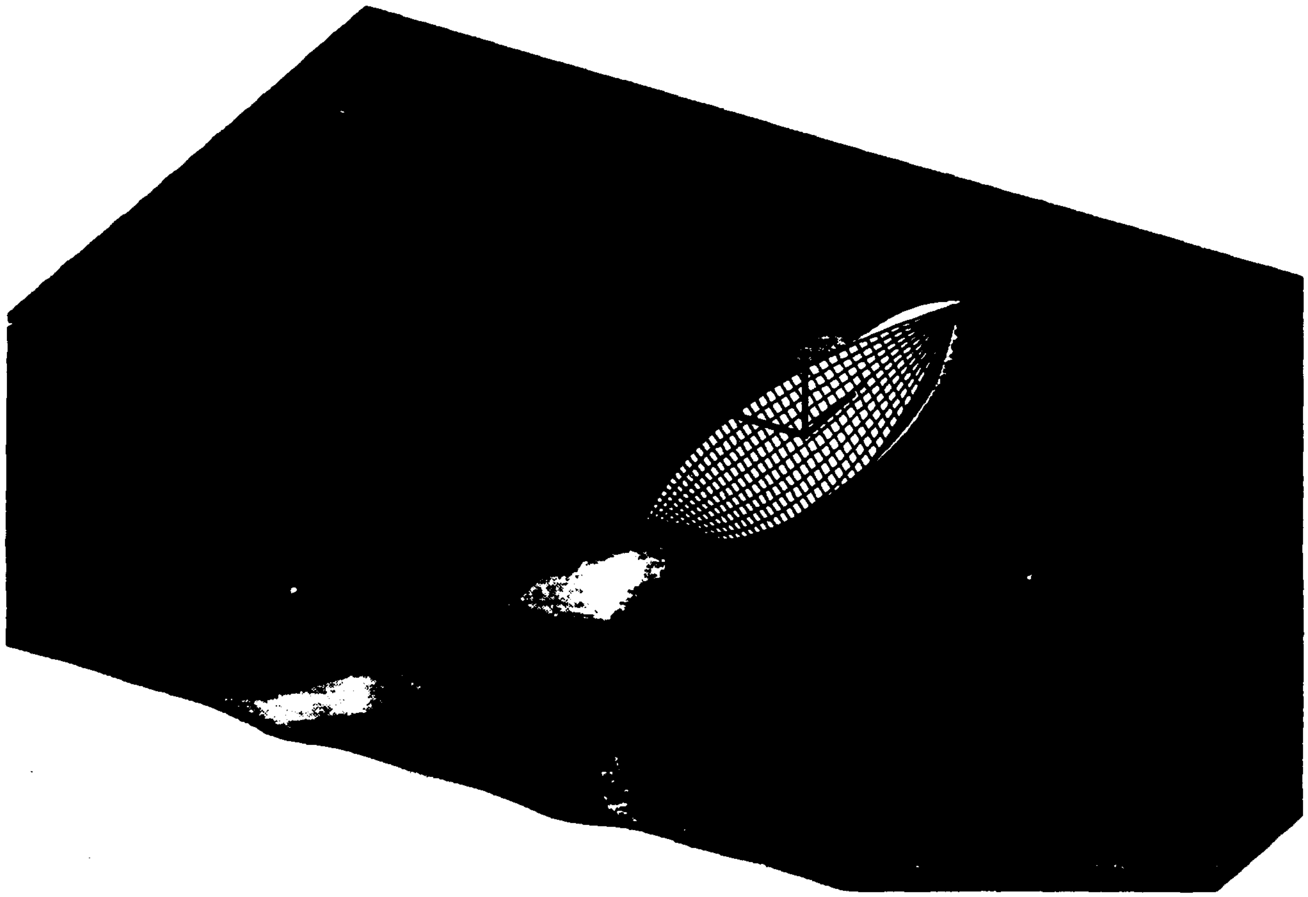


Figure 5-14: Steady wave pattern for a transom hull at $\mathcal{F} = 0.3$, viewed obliquely from above and behind the vessel.

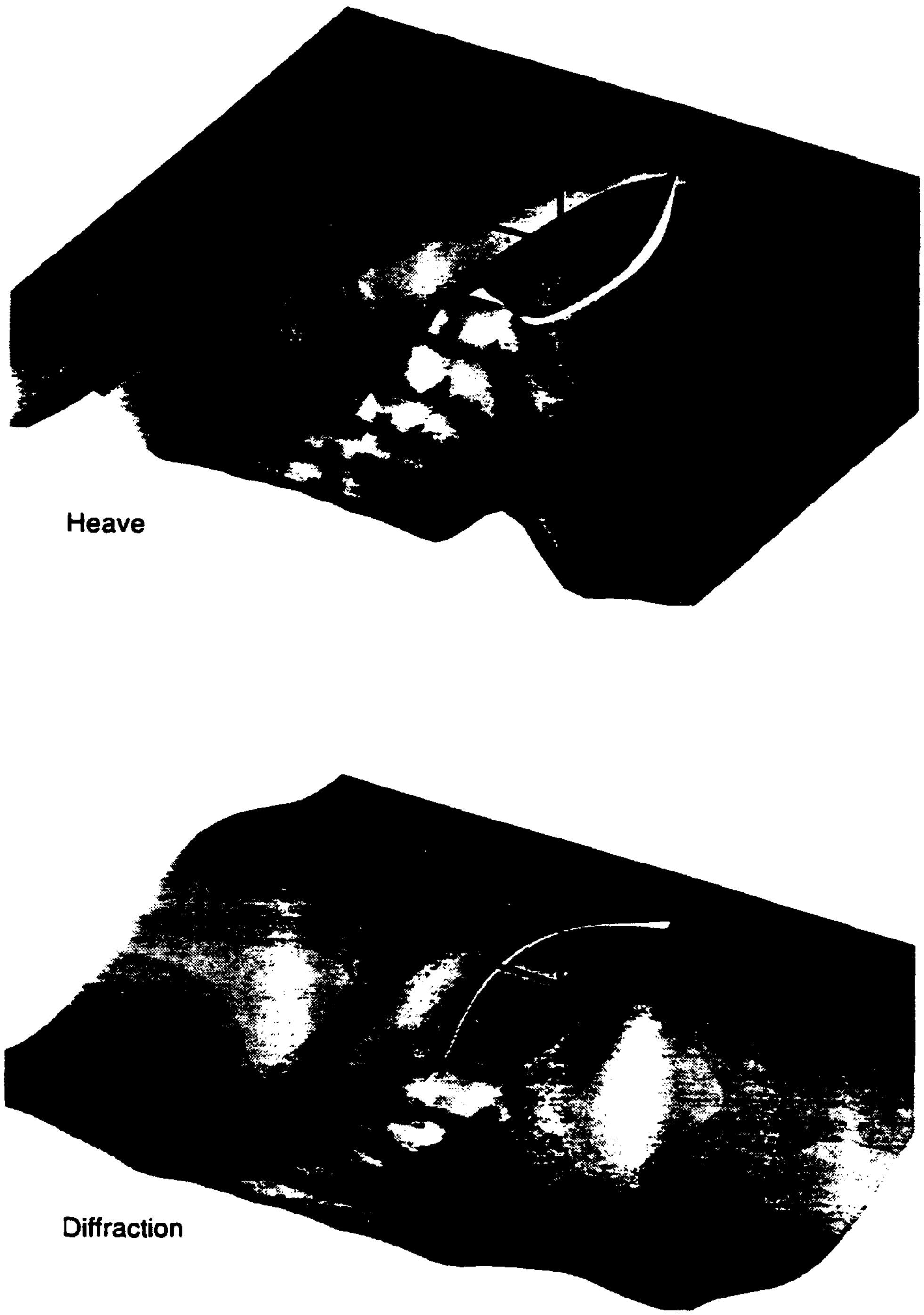


Figure 5-18: Heave and diffraction wave patterns and linear body pressure patterns for a transom hull at $\mathcal{F} = 0.3$ and encounter frequency $\omega/(g/L)^{\frac{1}{2}} = 3.2$.

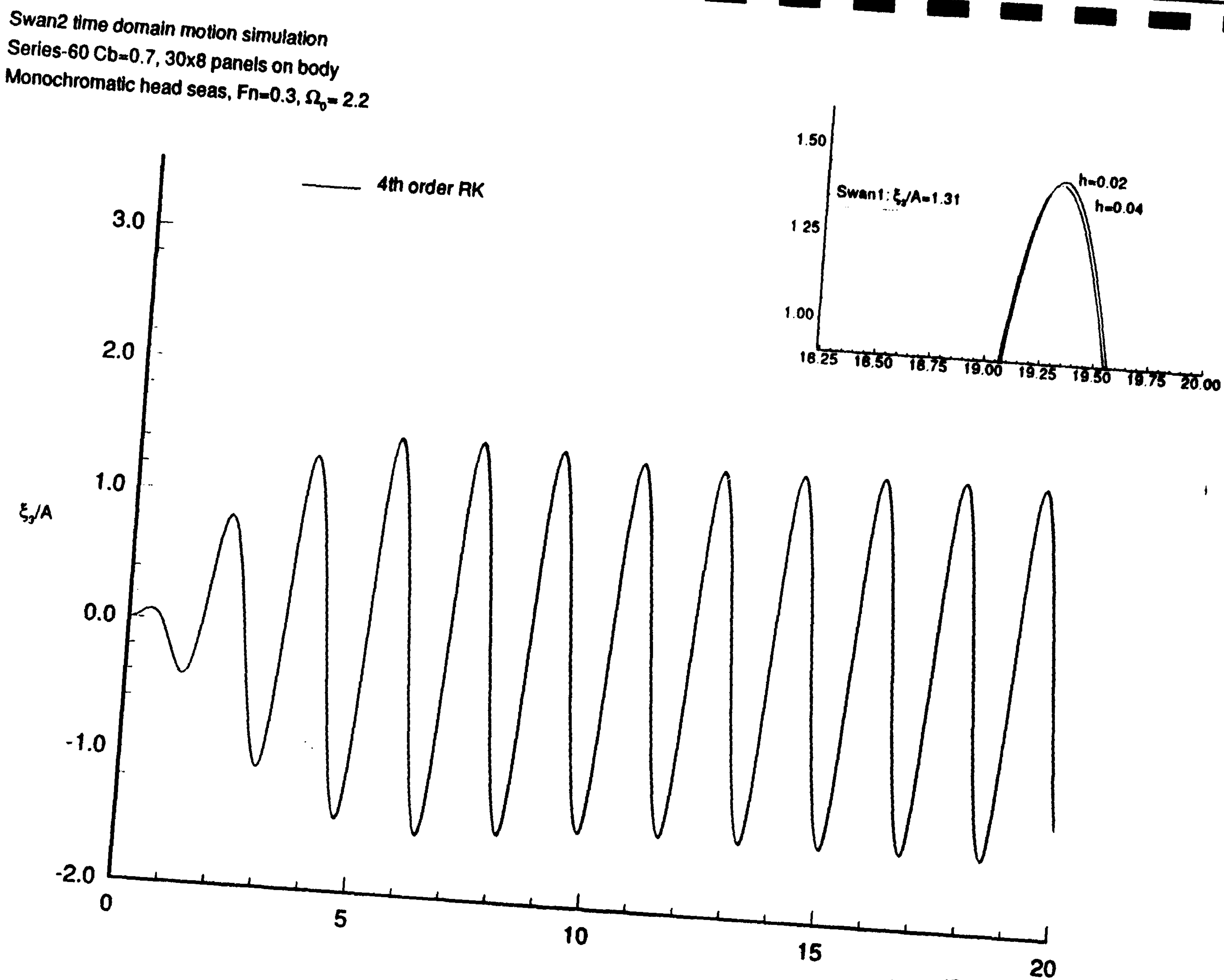
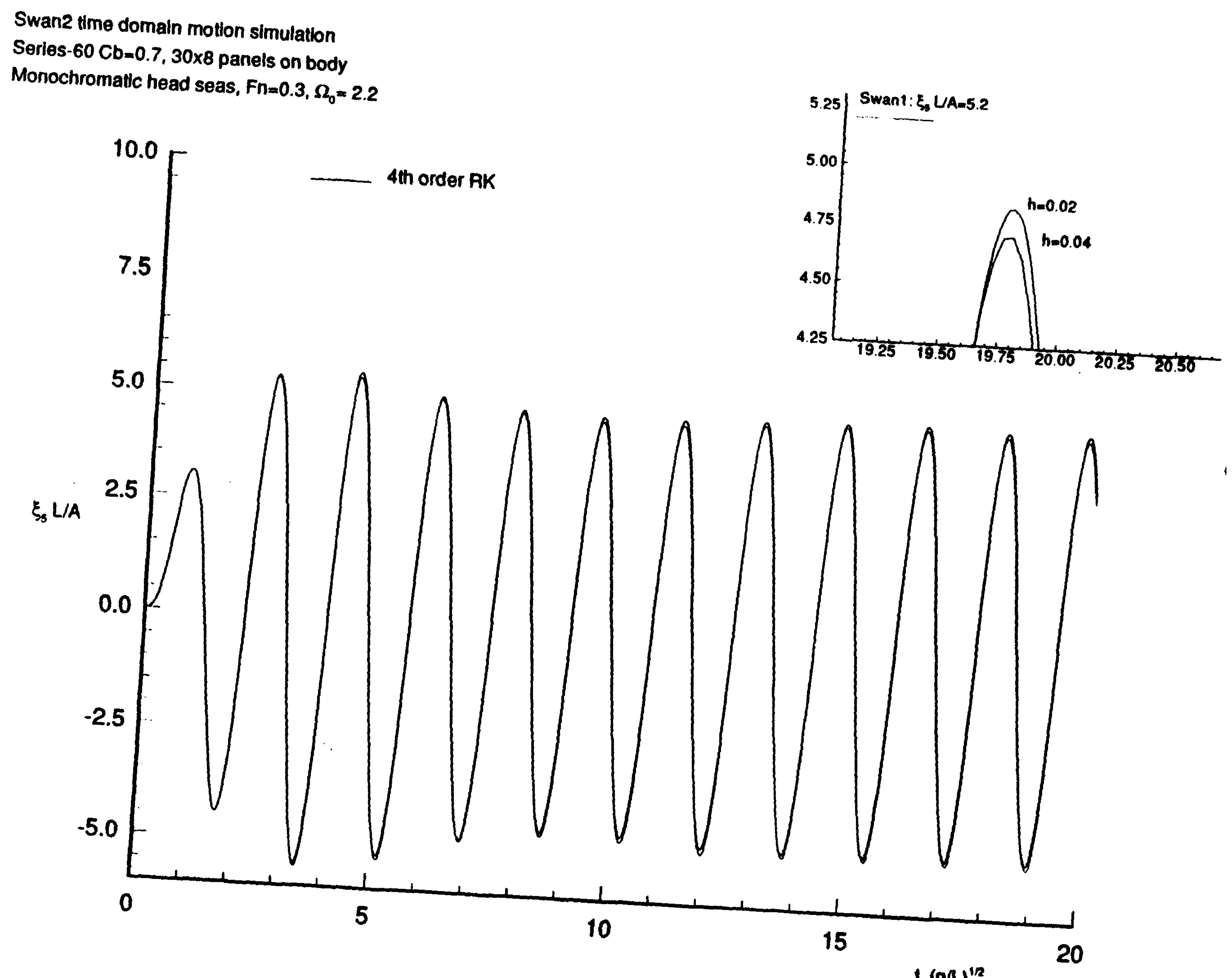

Swan2 time domain motion simulation
Series-60 Cb=0.7, 30x8 panels on body
Monochromatic head seas, Fn=0.3, Ω_0= 2.2
4th order RK
Swan1: ξ_3/A=1.31
h=0.02
h=0.04
1.50
1.25
1.00
18.25
18.50
18.75
19.00
19.25
19.50
19.75
20.00
3.0
2.0
1.0
0.0
-1.0
-2.0
ξ_3/A
0
5
10
15
20
t $(g/L)^{1/2}$

Figure 6



Swan2 time domain motion simulation
Series-60 Cb=0.7, 30x8 panels on body
Monochromatic head seas, Fn=0.3, Ω_0= 2.2
4th order RK
ξ_5 L/A
10.0
7.5
5.0
2.5
0.0
-2.5
-5.0
0
5
10
15
20
t $(g/L)^{1/2}$
Swan1: ξ_5 L/A=5.2
h=0.02
h=0.04
5.25
5.00
4.75
4.50
4.25
19.25
19.50
19.75
20.00
20.25
20.50

Figure 7

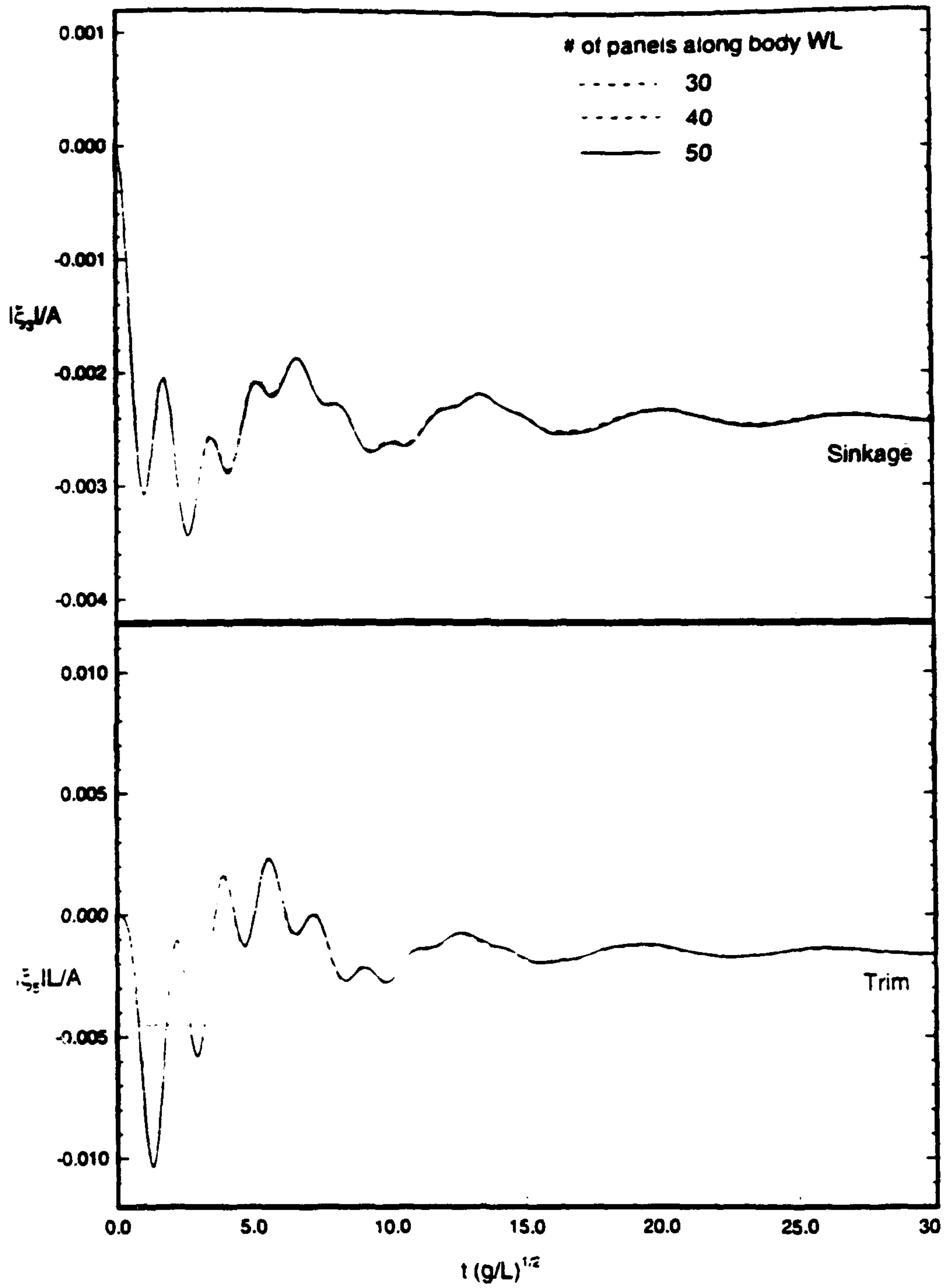


Figure 6-1: Convergence of heave and pitch motions with spatial discretization for the modified Wigley hull in the transition from rest to steady-state equilibrium position at $\mathcal{F} = 0.3$.

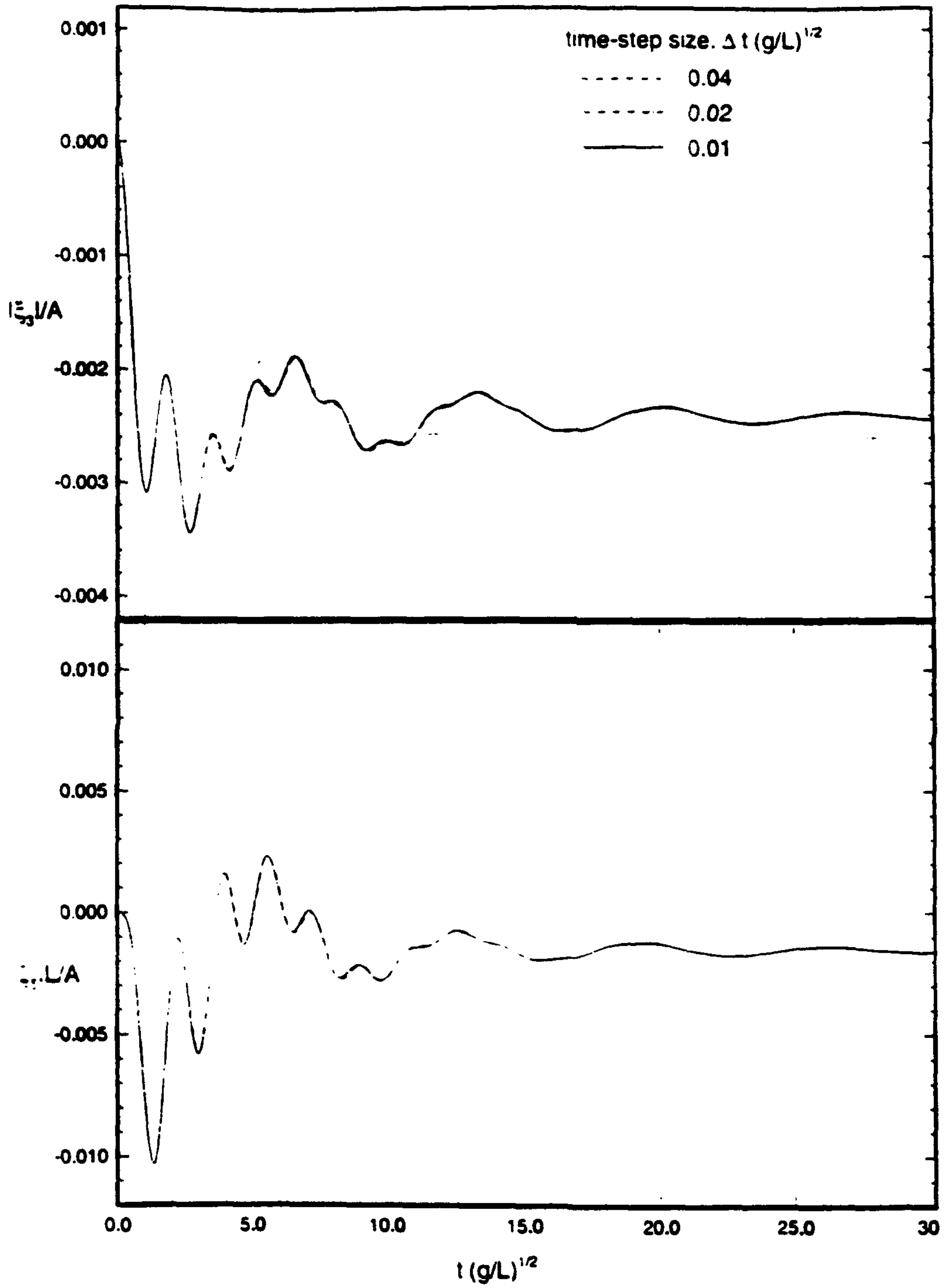


Figure 6-2: Convergence of heave and pitch motions with temporal discretization for the modified Wigley hull in the transition from rest to steady-state equilibrium position at $\mathcal{F} = 0.3$.

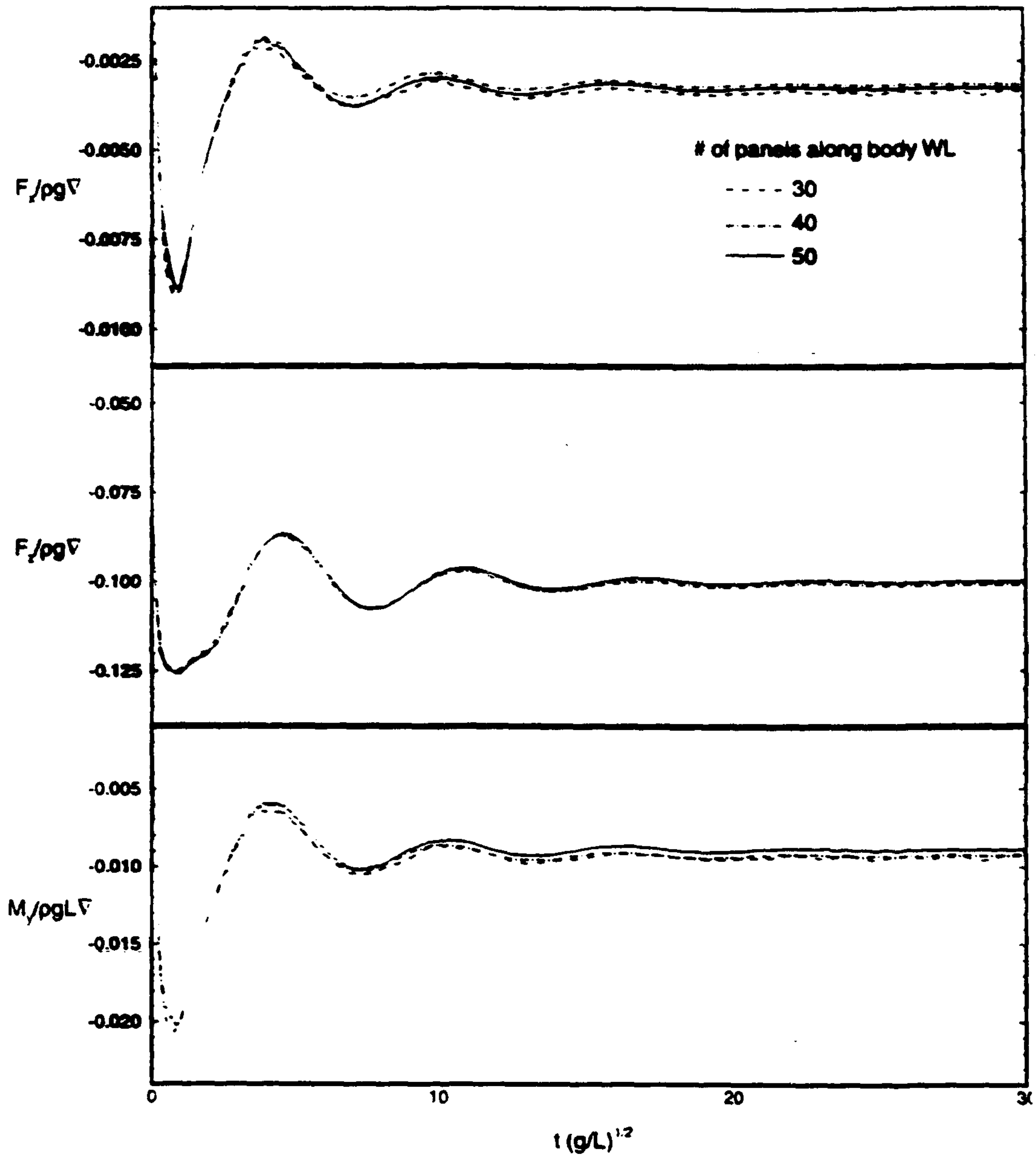


Figure 5-16: Convergence of forces with spatial discretization for a transom hull in steady forward motion at $\mathcal{F} = 0.3$.

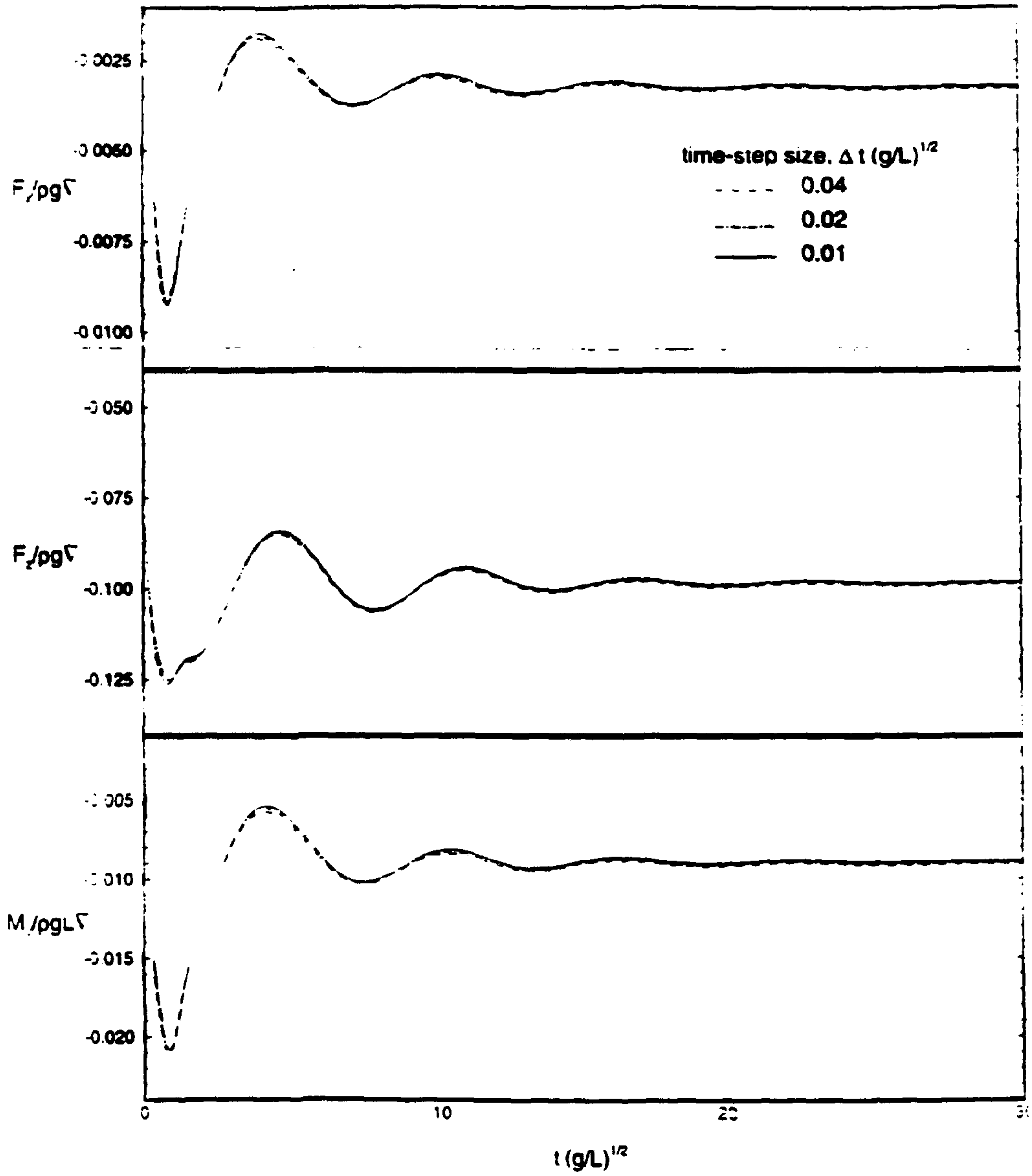


Figure 5-17: Convergence of forces with temporal discretization for a transom hull in steady forward motion at $\mathcal{F} = 0.3$.

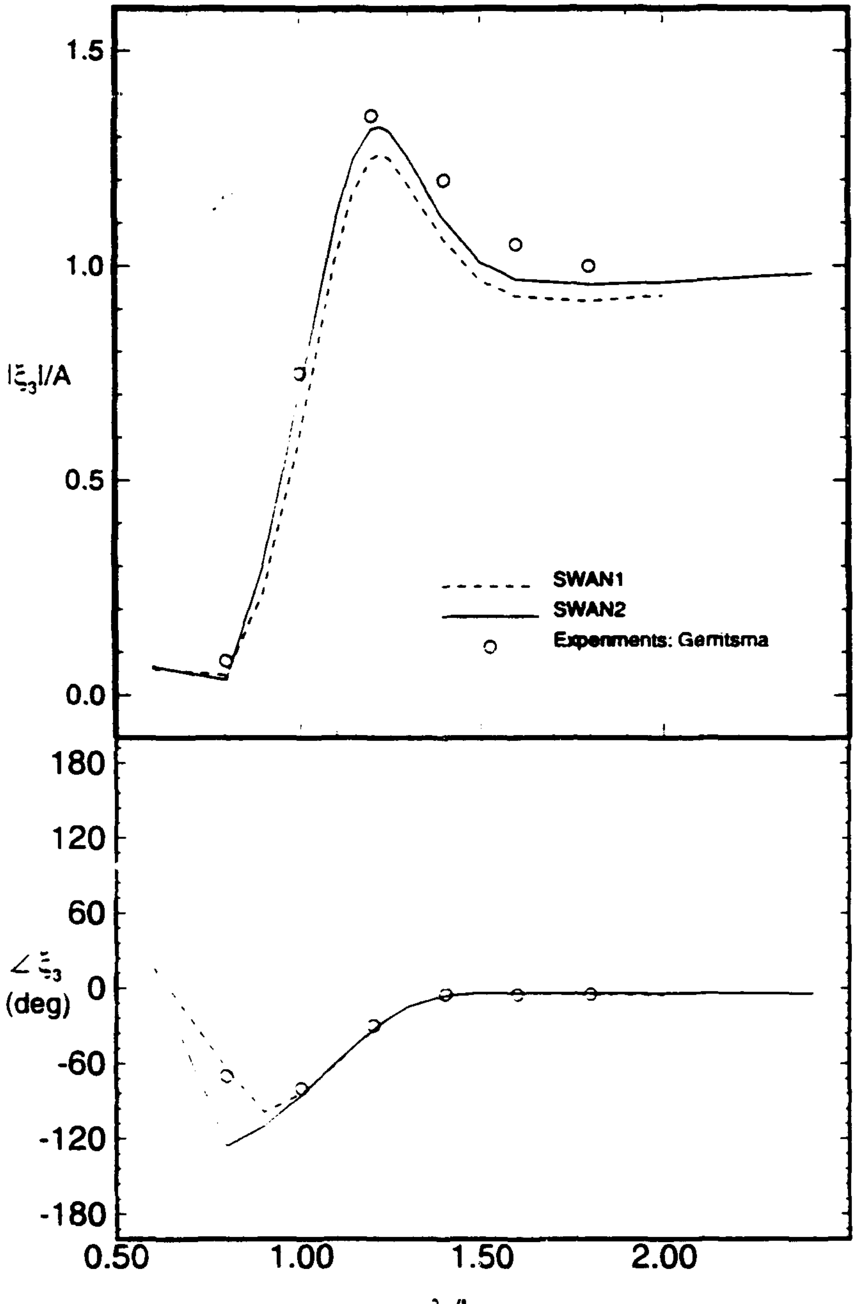


Figure 6-7: Magnitude and phase of the heave response amplitude operator for the Series 60 at $\mathcal{F} = 0.2$.

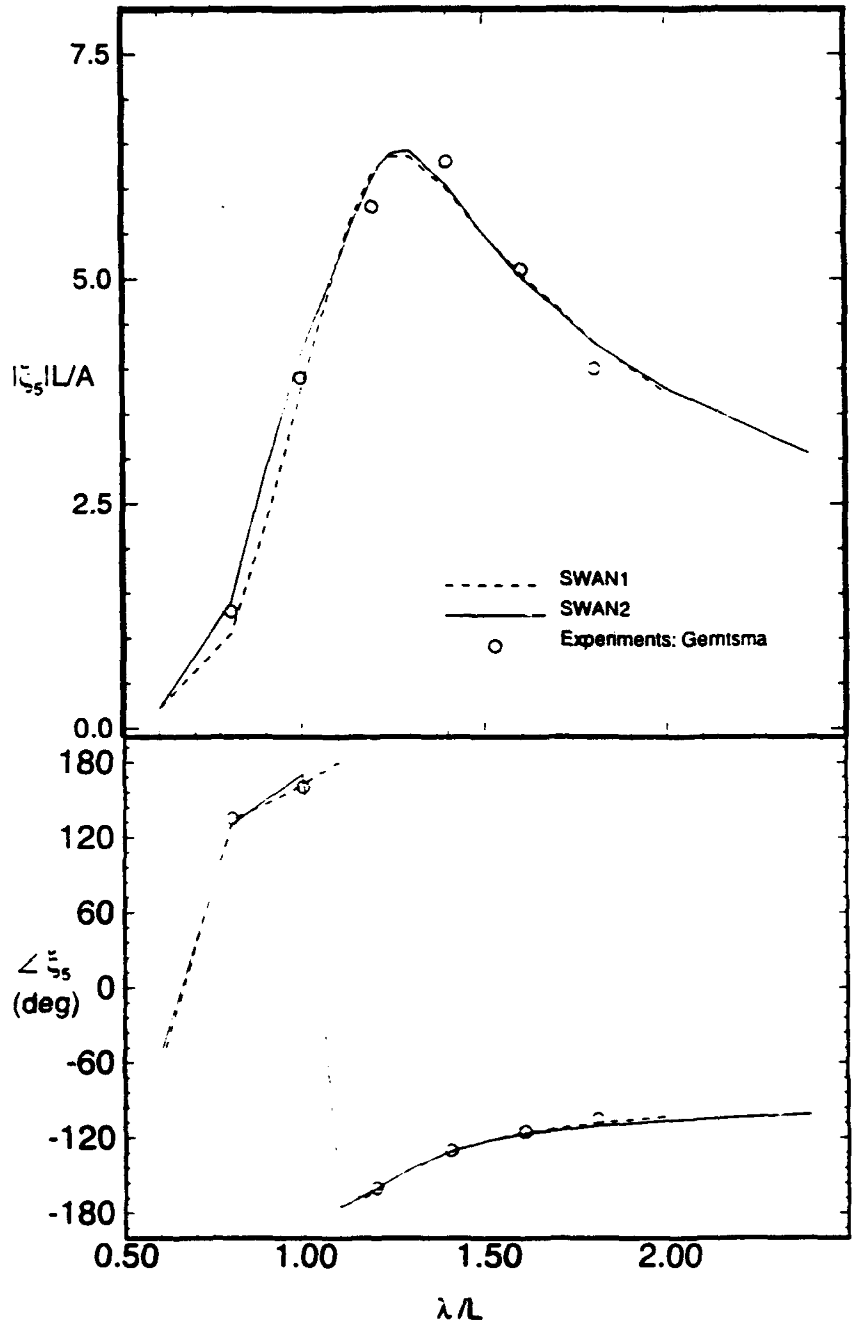


Figure 6-8: Magnitude and phase of the pitch response amplitude operator for the Series 60 at $\mathcal{F} = 0.2$.

PREDICTION OF NONLINEAR LOADING OF FLARED BODIES USING A NUMERICAL TOWING TANK

Presented by

B. MASKEW
ANALYTICAL METHODS, INC.
REDMOND, WASHINGTON

Presented at

THE ONR WORKSHOP ON NONLINEAR SEA LOADS AND SHIP RESPONSE:
A BASIS FOR SHIP STRUCTURAL DESIGN

COLLEGE OF ENGINEERING, UNIVERSITY OF MICHIGAN
ANN ARBOR, MICHIGAN
JULY 7TH AND 8TH, 1994

OBJECTIVES

GENERAL PURPOSE TIME-DOMAIN COMPUTATIONAL TOOL FOR FULLY NON-LINEAR SIMULATIONS OF 3-D SHIP HYDRODYNAMICS

- NON-LINEAR SEAKEEPING
- MANEUVER EVENTS
- LARGE AMPLITUDE MOTIONS
- LOADS AND LOAD DISTRIBUTIONS
- SLAMMING LOADS AND WAVE IMPACT EFFECTS
- DECK WETNESS
- COUPLING WITH STRUCTURES CODE

ISSUES

- CONTACT LINE DYNAMICS — REPANELLING
- JET FORMATION — IDENTIFICATION AND MODELING
- GREEN WATER ON DECK
- ROLL DAMPING MODEL — TRANSIENT SEPARATION LINE TREATMENT
  — VORTEX CONVECTION
- CODE OPTIMIZATION AND PARALLELIZATION
- TANK BOUNDARY EFFECTS

METHOD

GENERAL PURPOSE UNSTEADY COMPUTER PROGRAM, USAERO, PLUS COUPLED MODULES:

FPI — "FLIGHT PATH" INTEGRATOR

FSP — FREE SURFACE PROGRAM

USAERO

- SURFACE SINGULARITY PANEL METHOD
- UNIFORM STRENGTH DOUBLET AND SOURCE PANELS
- TIME-STEPPING PROCEDURE
- MULTIPLE MOVING FRAMES OF REFERENCE

COUPLED ROUTINES FOR:-

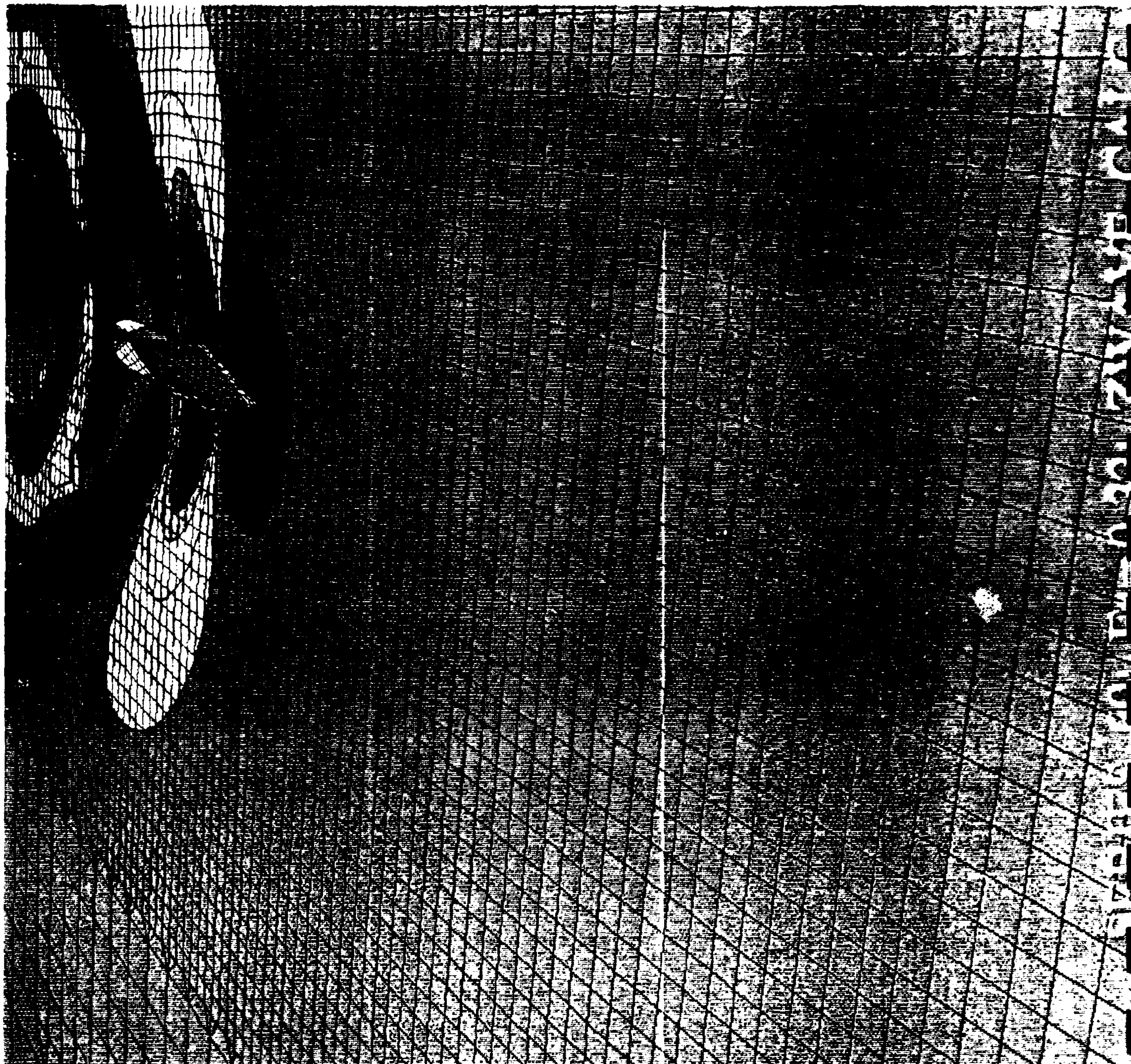
- VORTEX WAKE CONVECTION (LIFTING EFFECTS)
- BOUNDARY LAYER ANALYSIS (VISCOUS EFFECTS FOR DISPLACEMENT AND SKIN FRICTION

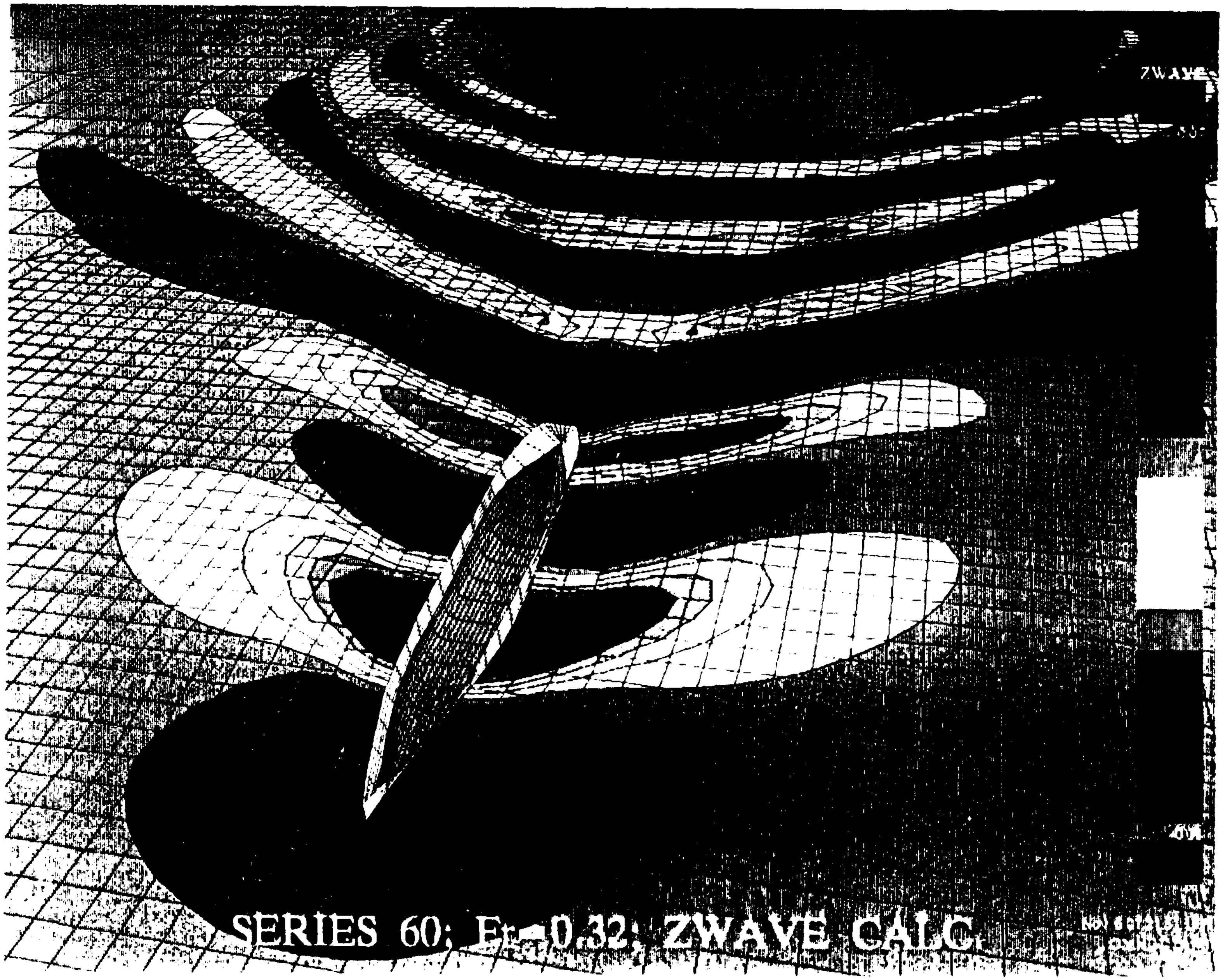
FPI MODULE

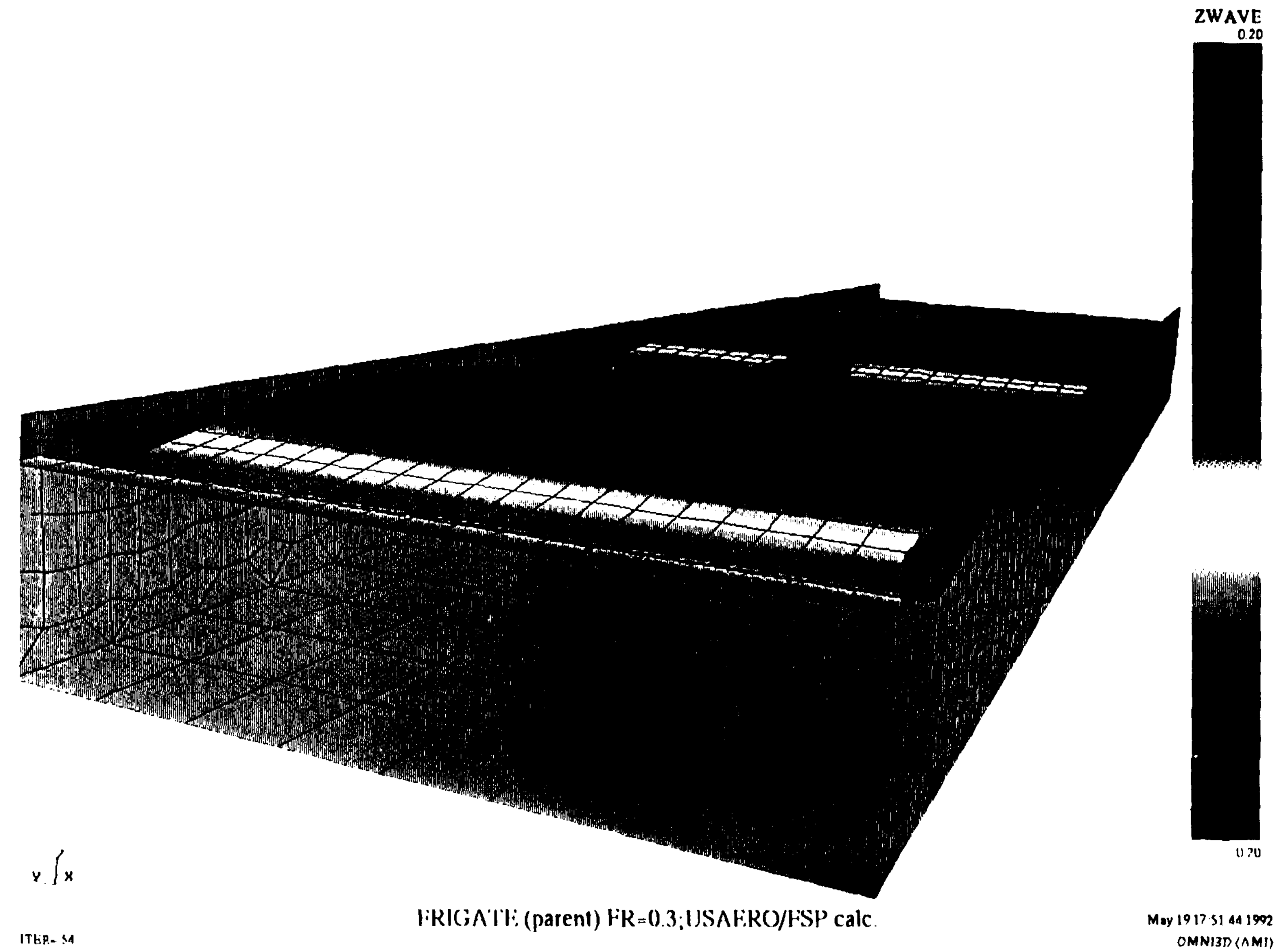
- DRIVEN BY INSTANTANEOUS FORCE AND MOMENT
- PROVIDES BODY LOCATION/ORIENTATION AND VELOCITY FOR NEXT TIME STEP
- CHOICE OF DEGREES OF FREEDOM UP TO 6
- VARIOUS INTEGRATION OPTIONS UP TO ADAMS BASHFORTH ORDER 5
- PROVISION FOR AUTOPILOT

FSP MODULE

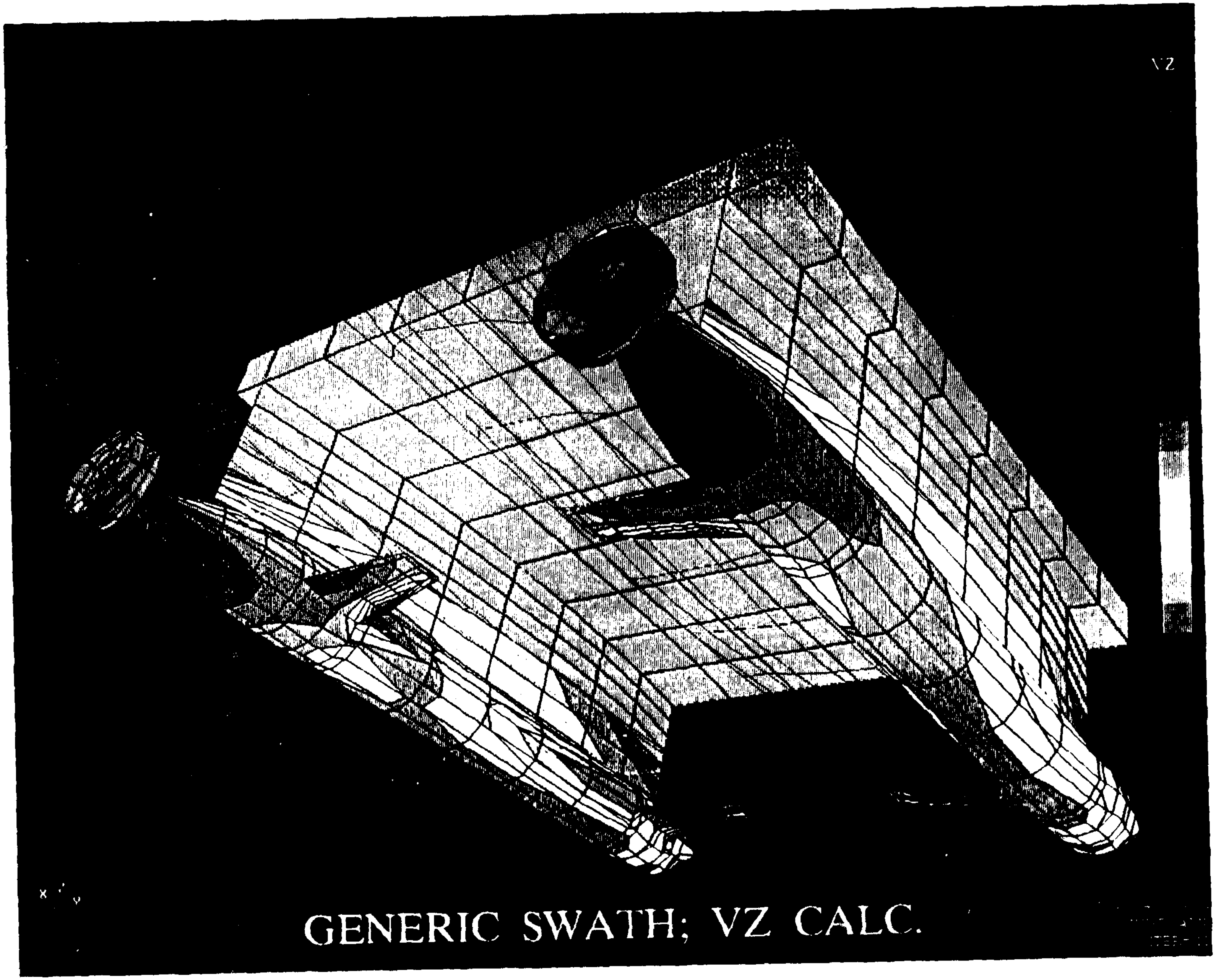
- NON-LINEAR TIME-DOMAIN TREATMENT
- DEFORMED FREE SURFACE
- AUTOMATIC TREATMENT OF FREE SURFACE/HULL INTER-SECTION (REPANELS)
- INCLUDES WAVE MAKER
- EXTENDED FOR "DRY" TRANSOM HULL TREATMENT




SERIES 60; Fr=0.32; ZWAVE CALC.



FRIGATE (parent) FR=0.3;USAERO/FSP calc.


VZ
GENERIC SWATH; VZ CALC.

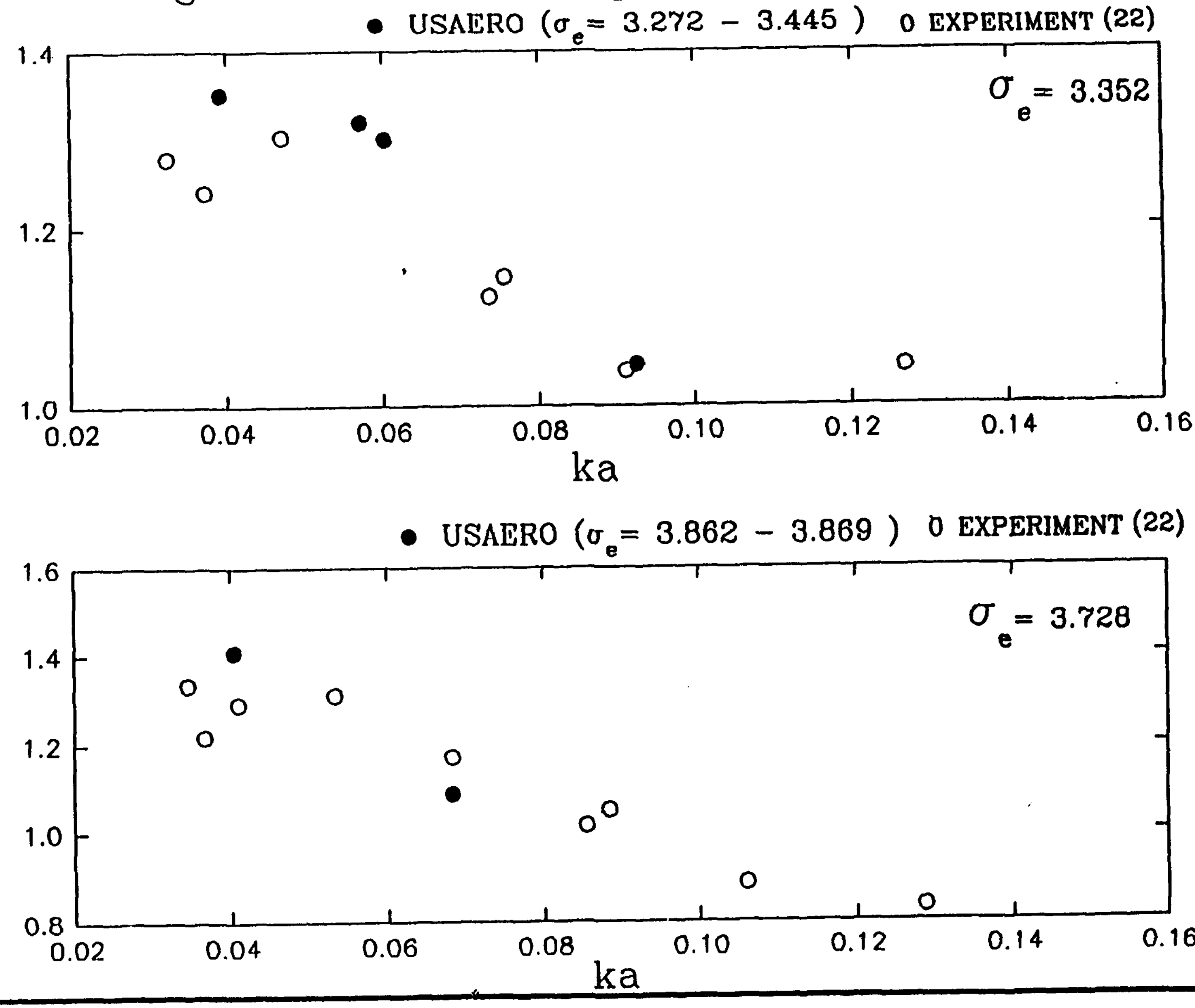
S175 ITTC HULL FORM

EXPERIMENTS—REF. O'DEA, POWERS, ZSELECSKY, 1992

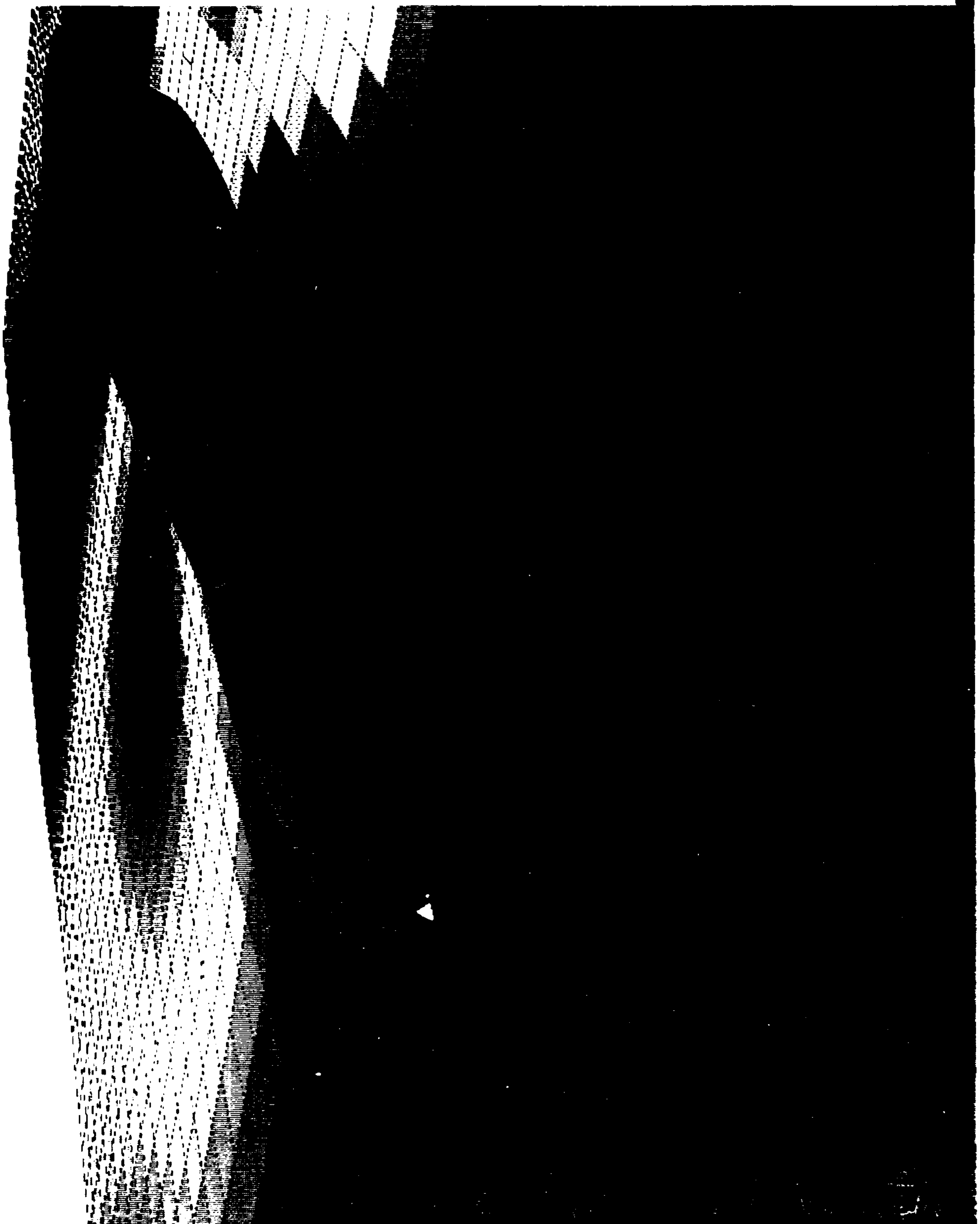
FROUDE	0.275
LENGTH (Lpp)	3.5m
BEAM	0.51m
DRAFT	0.19m
C_B	0.572
Ryy/Lpp	0.24
DISPLACEMENT	193.2kg

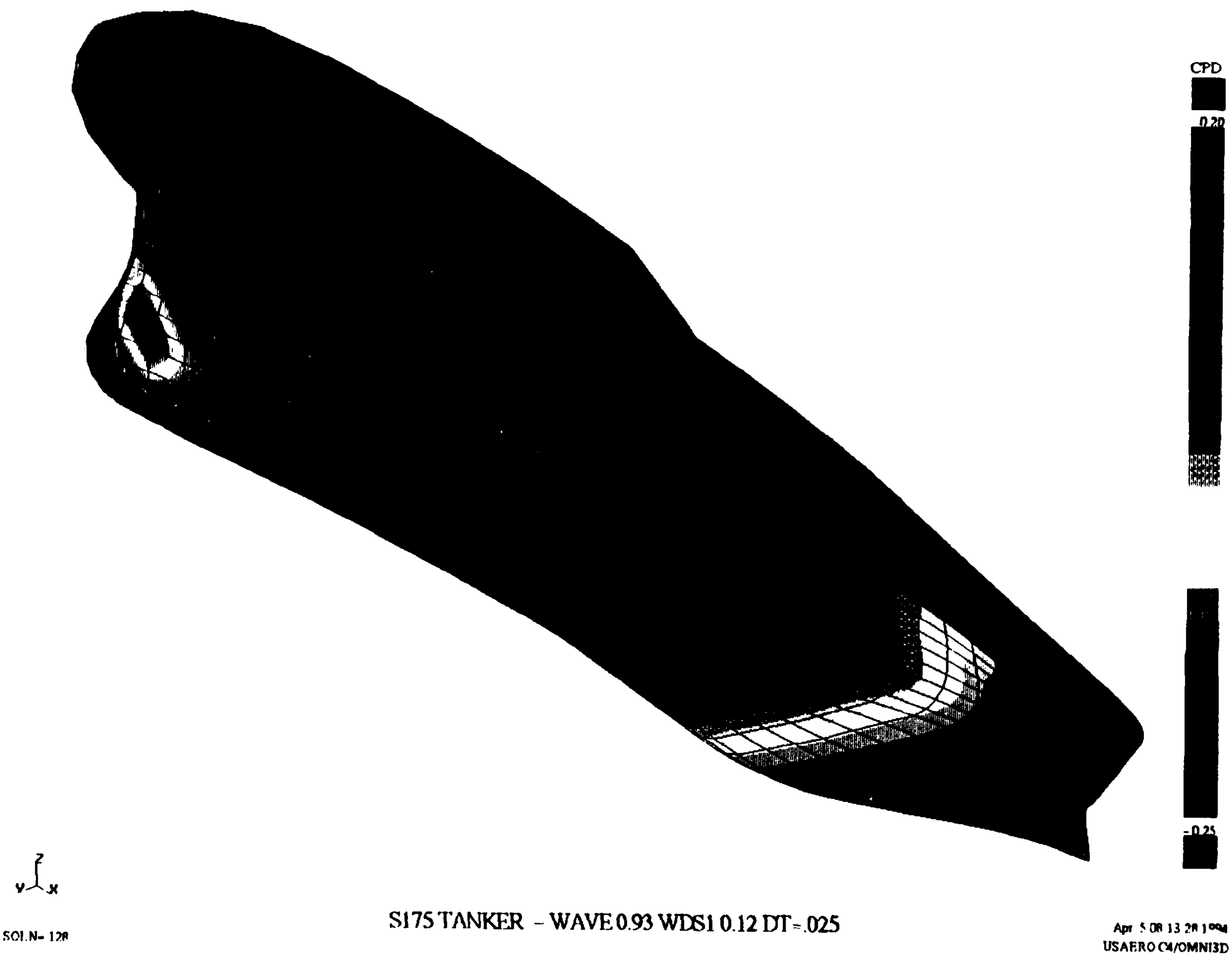
S175 TANKER

Magnitude of heave response function Fn=0.275

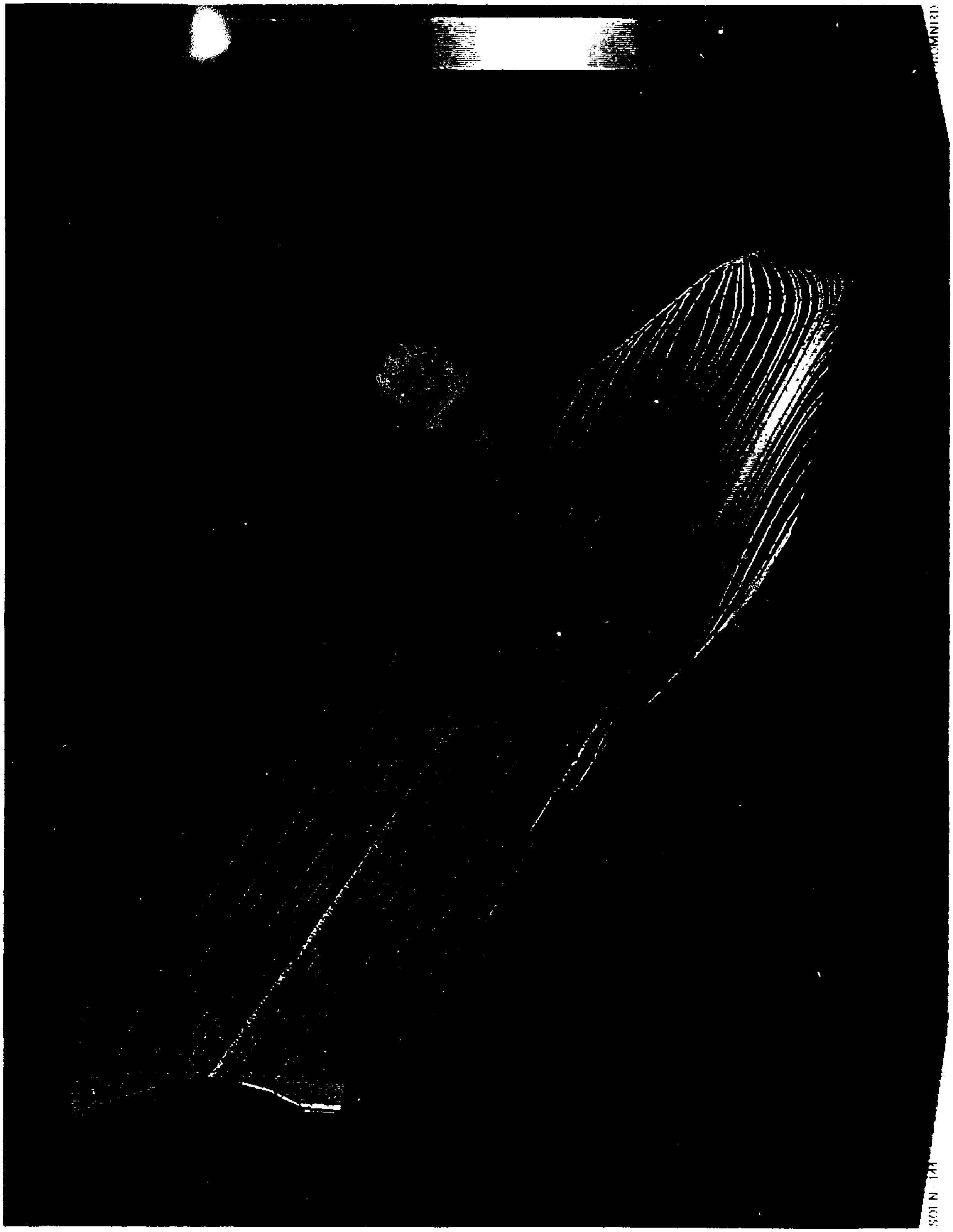


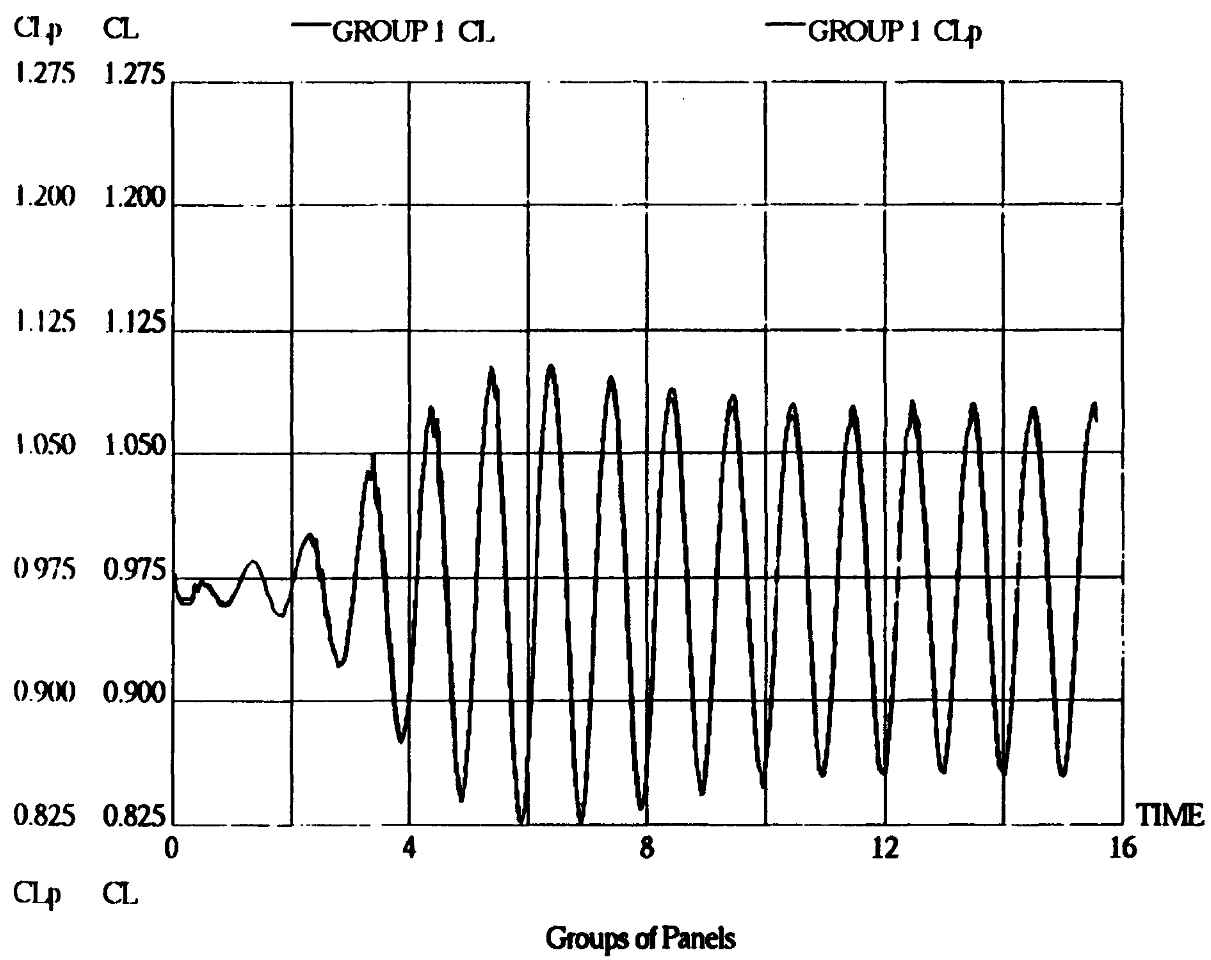
AMT





S175 TANKER - WAVE 0.93 WDS1 0.12 DT=.025

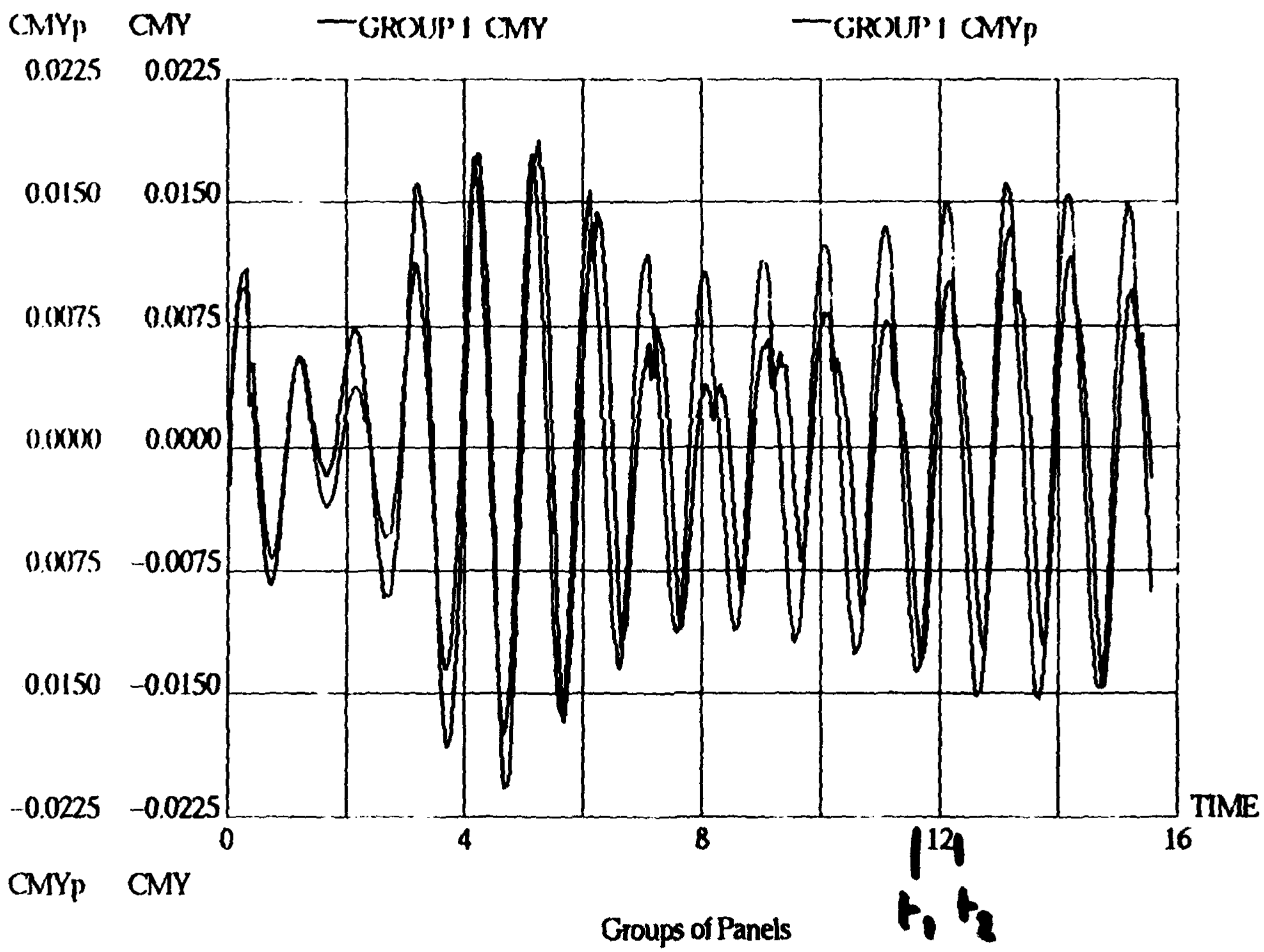




S175 CONTAINER SHIP - WAVE 0.93 WDS1 0.06 DT=.03

SOLN- 8

Jul 6 07:34 08 1994
USAERO C4/OMNI3D

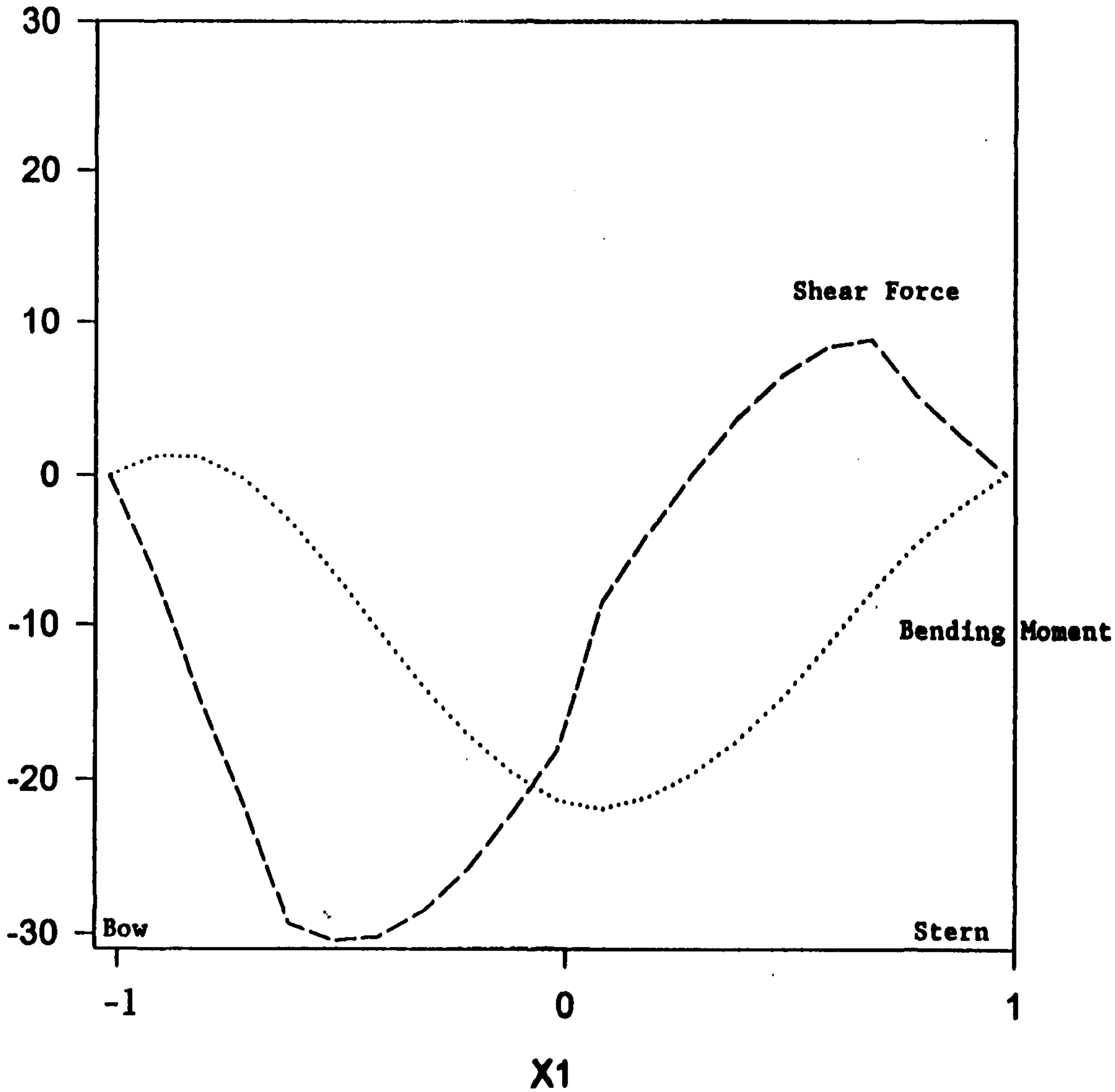


S175 CONTAINER SHIP - WAVE 0.93 WDS1 0.06 DT=.03

SOLN= 8

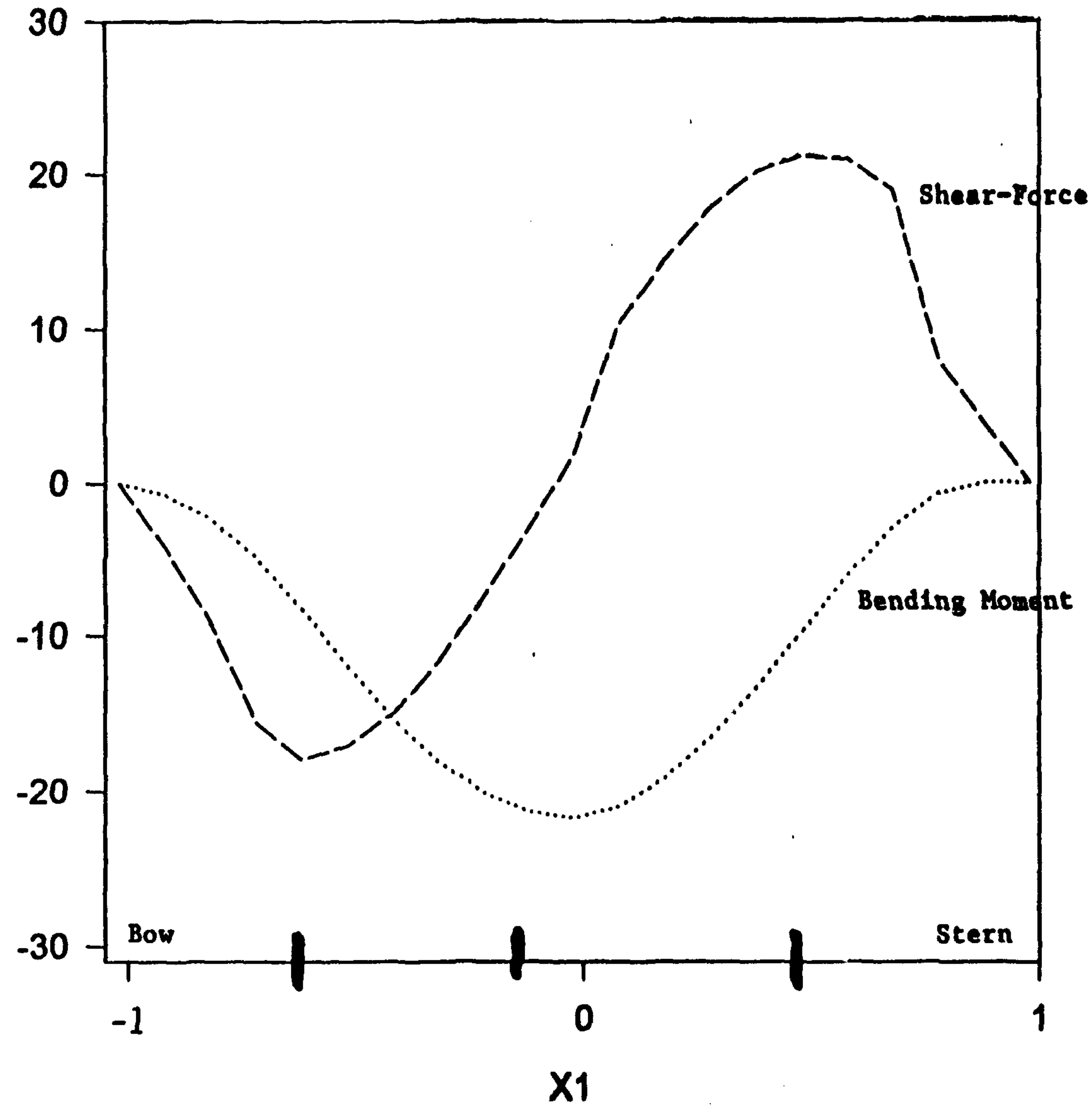
Jul 6 07:31:06 1994
USAERO C4/OMNI3D

time t_1 = 11.64 sec

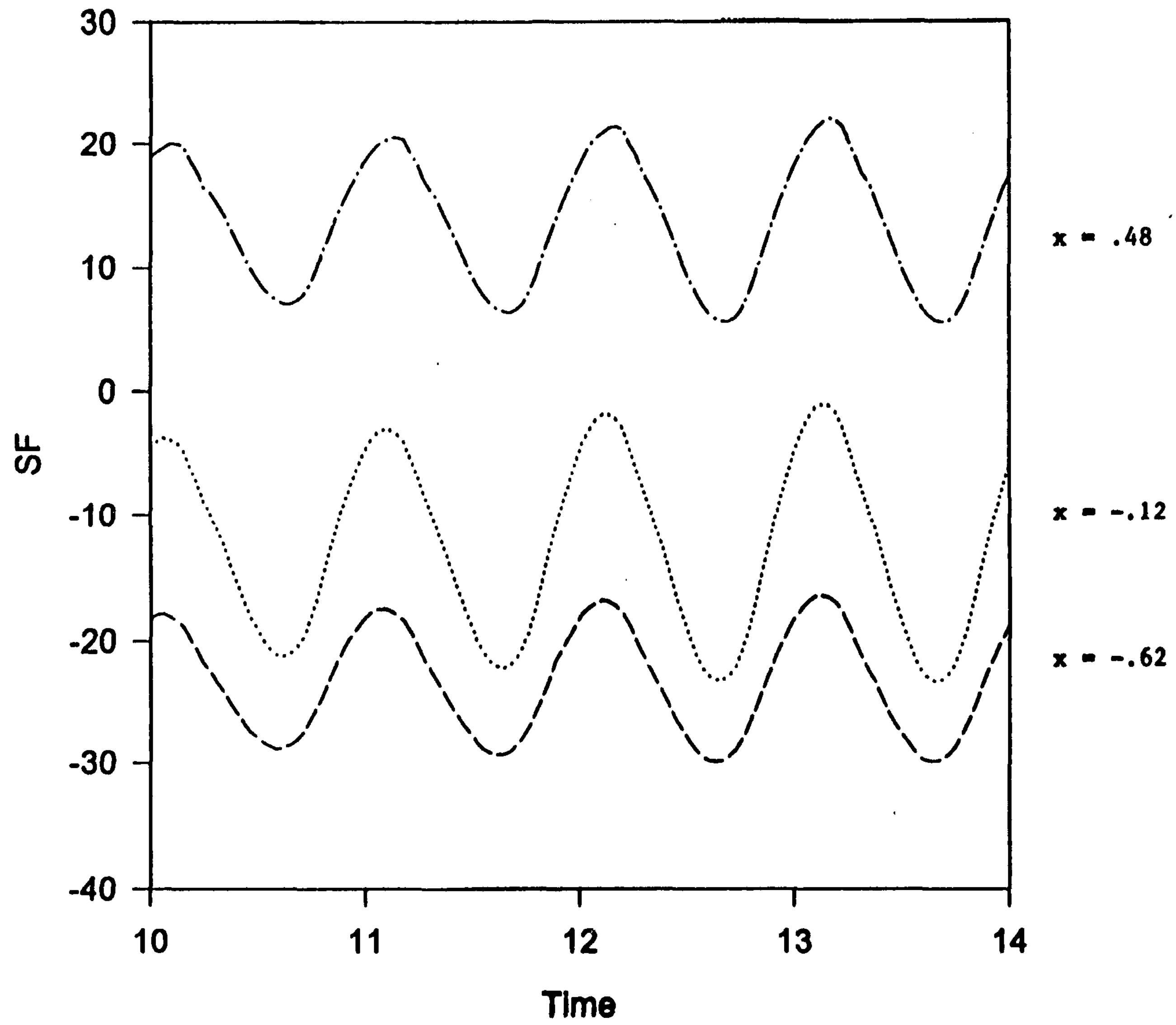


S175 Container Ship
Shear-Force and Bending Moment Distribution

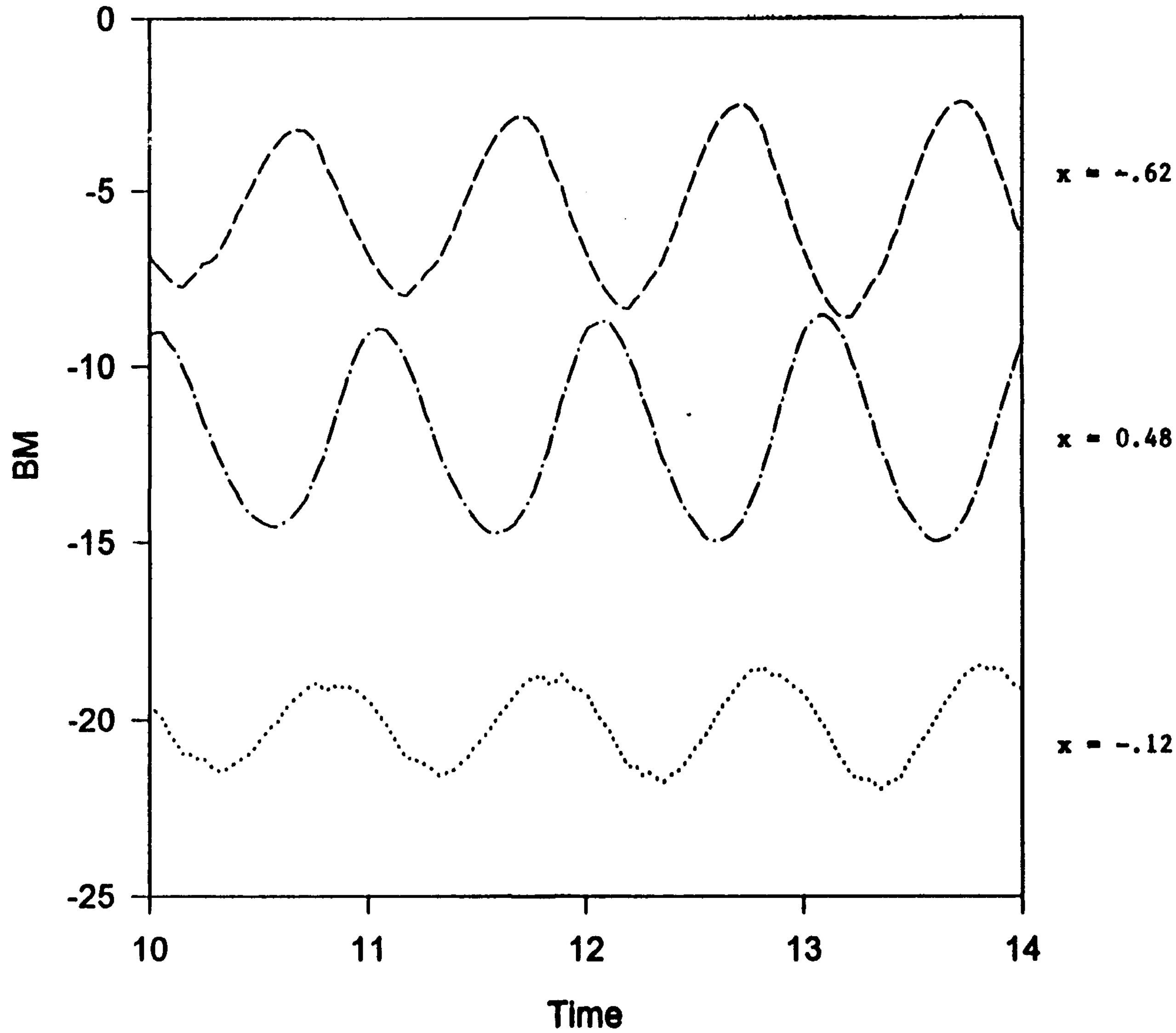
time T_z = 12.2 sec



S175 Container Ship
Shear-Force and Bending Moment Distribution



S175 Container Ship
Shear-Force History at Three Stations

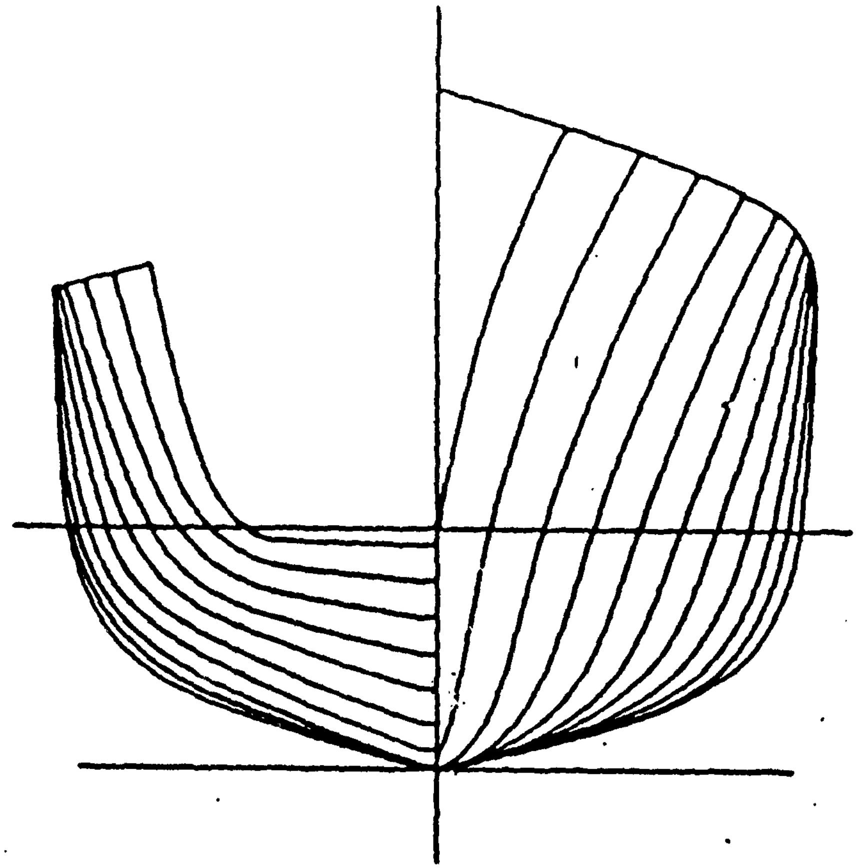


S175 Container Ship
Bending Moment History at Three Stations

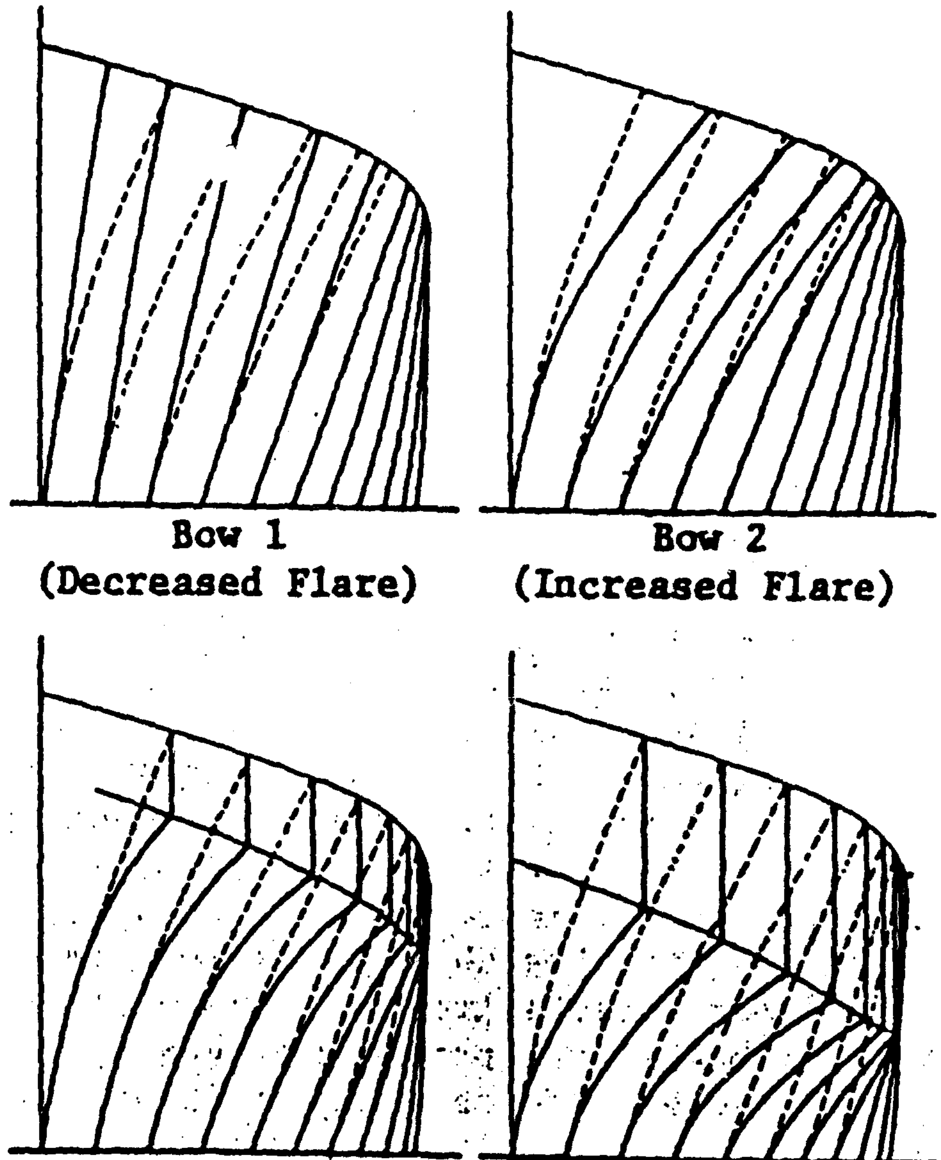
FRIGATE MODEL

EXPERIMENTS—REF. O'DEA & WALDEN, 1984

FROUDE	0.3
MODEL LENGTH, Lpp	4.5m
BEAM	0.496m
DRAFT	0.163m
C_B	0.454
Ryy/Lpp	0.277
DISPLACEMENT	165kg



Model Length - 4.5m
Beam - 0.496 m
Draft - 0.163 m
C_B - 0.454
C_x - 0.755



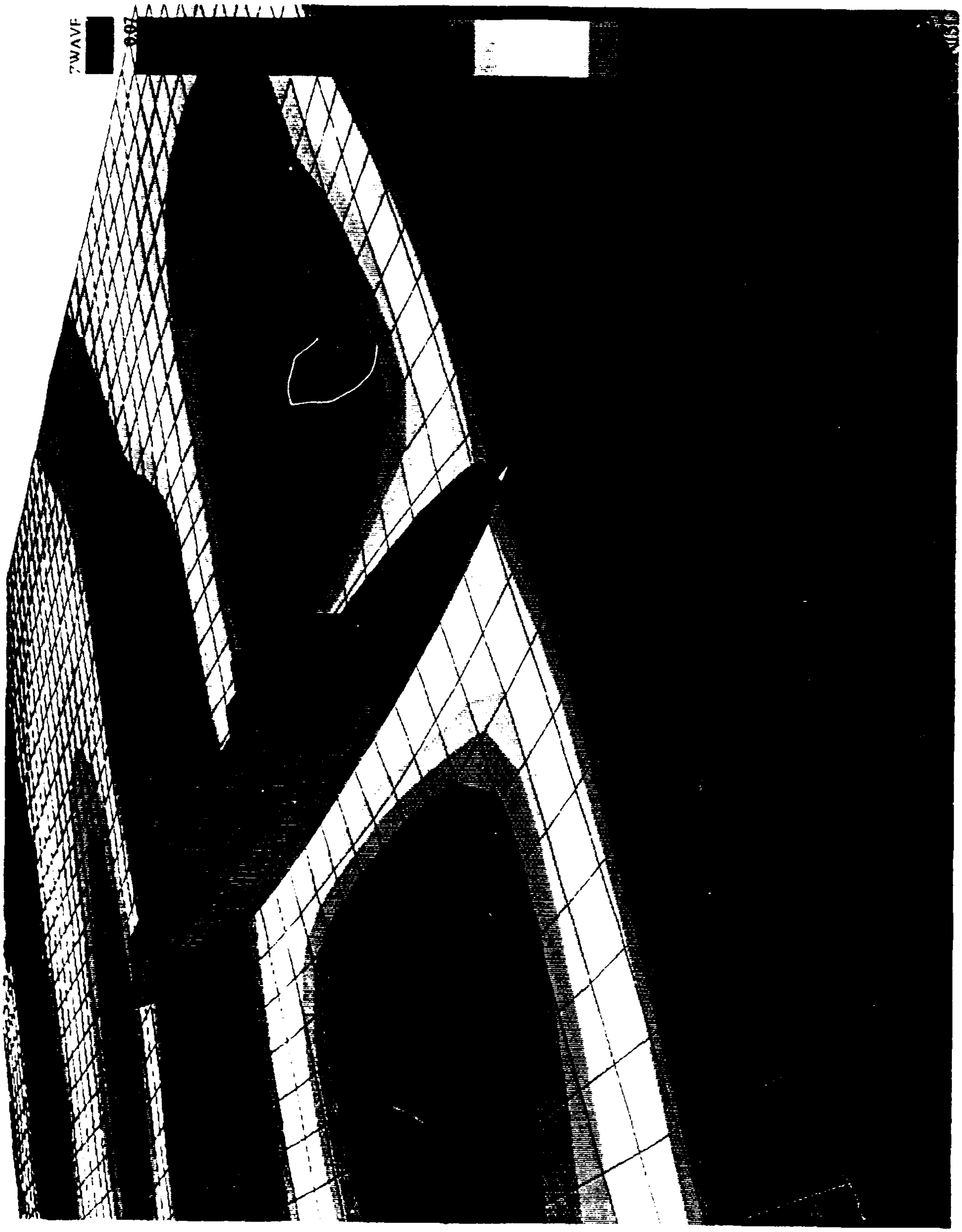
Bow 1
(Decreased Flare)

Bow 2
(Increased Flare)

Bow 3
(Shallow Knuckle)

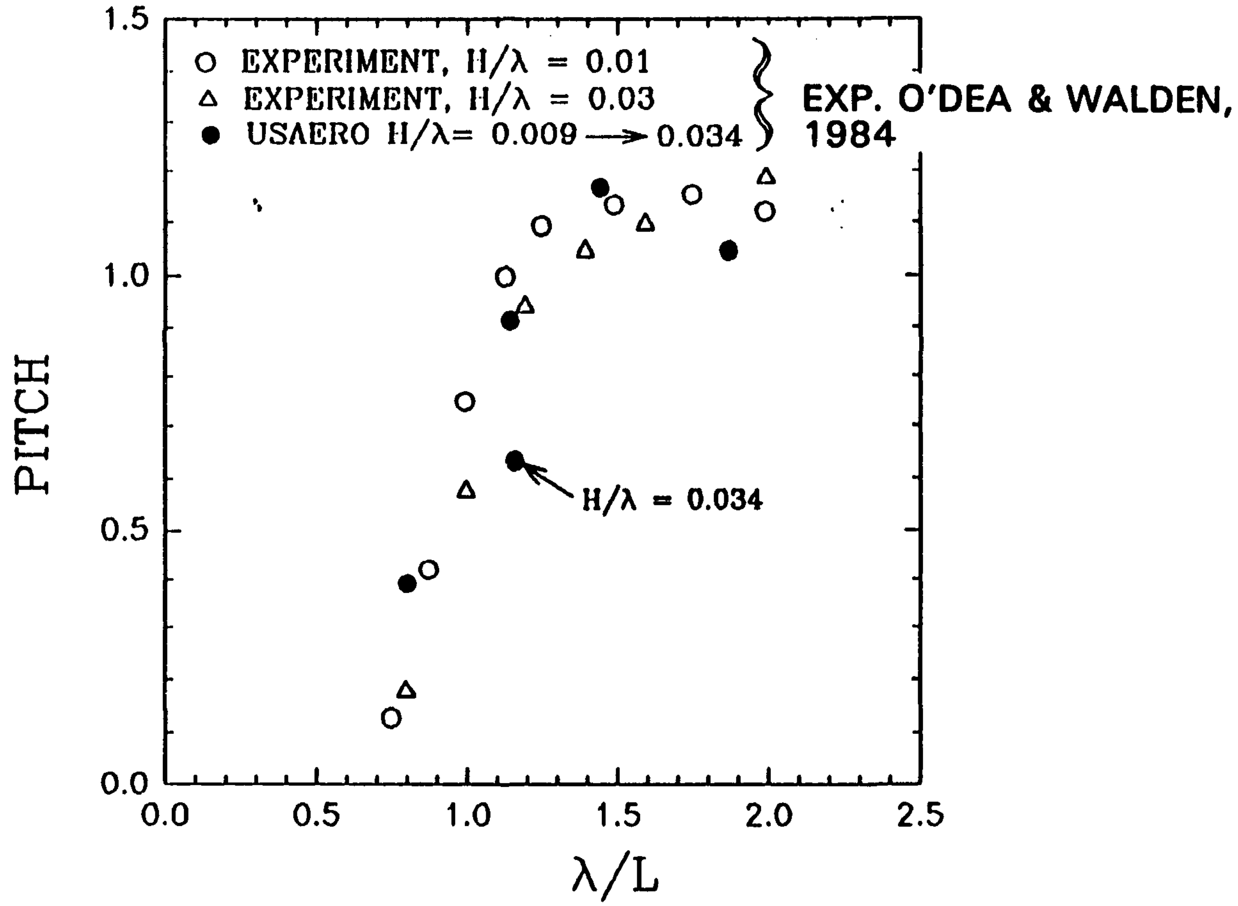
Bow 4
(Deep Knuckle)

Figure 1 - Model Principal Characteristics


ZWAVE
0.07

FRIGATE

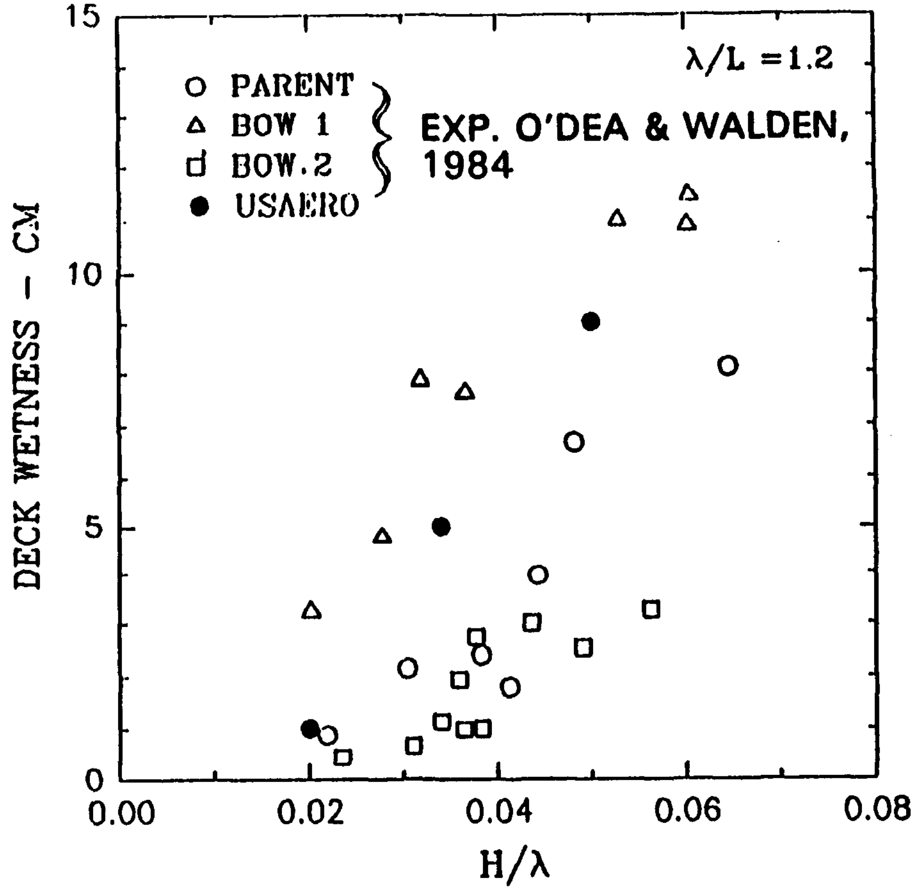
PITCH TRANSFER FUNCTION (Fn=0.30)



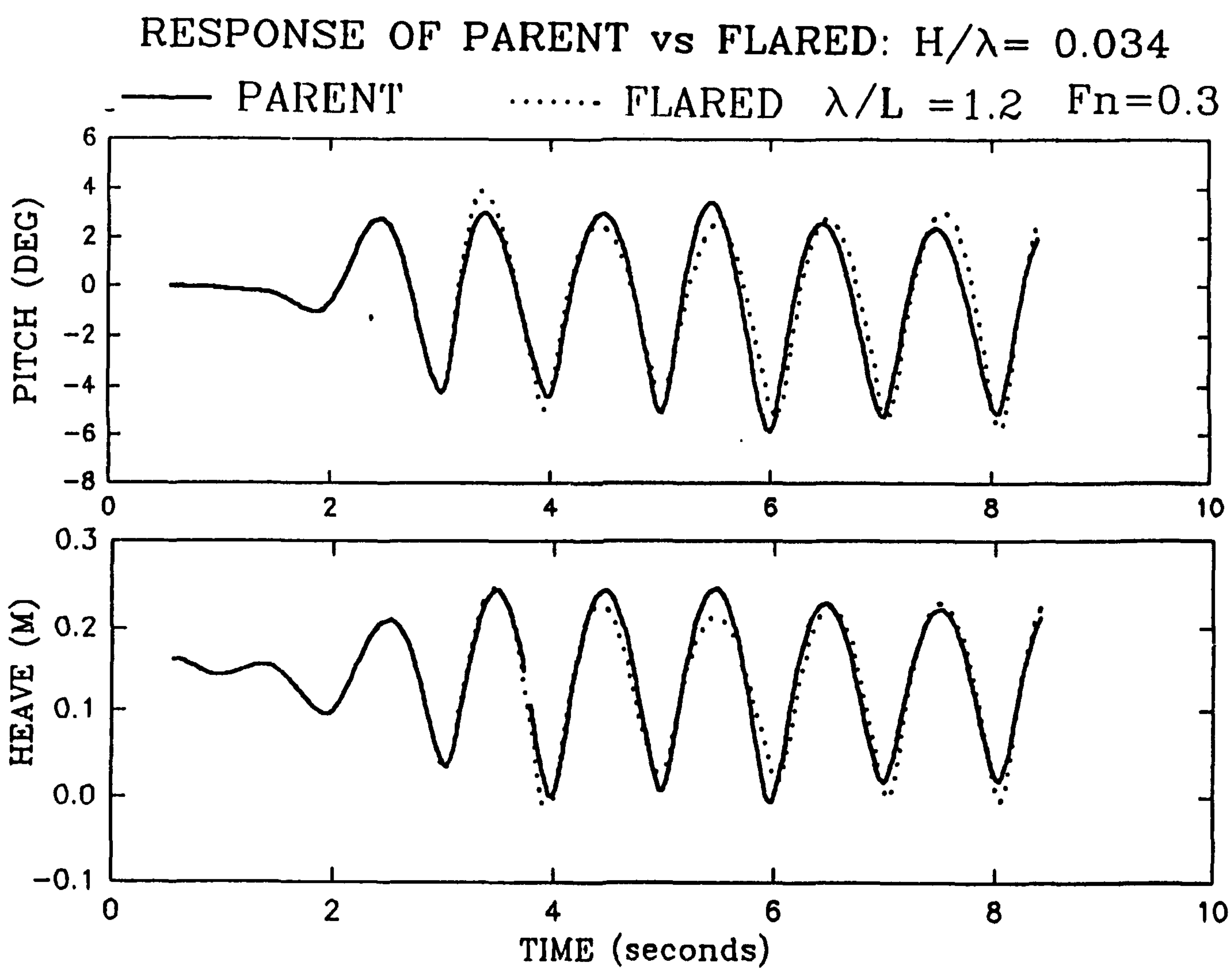
AMI

FRIGATE

VARIATION OF DECK WETNESS WITH WAVE STEEPNESS, DATA vs USAERO/FSP/FPI

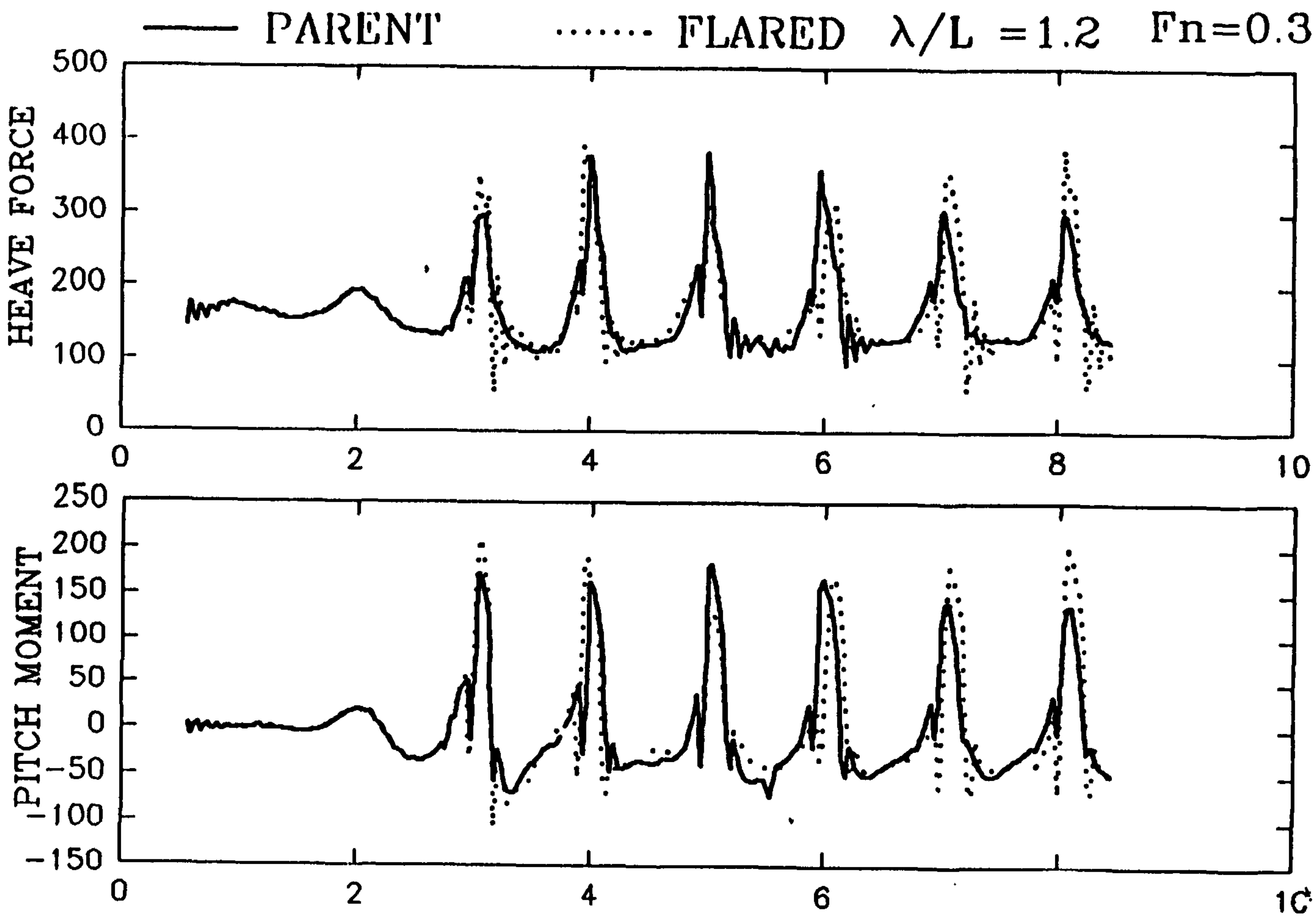


AMI

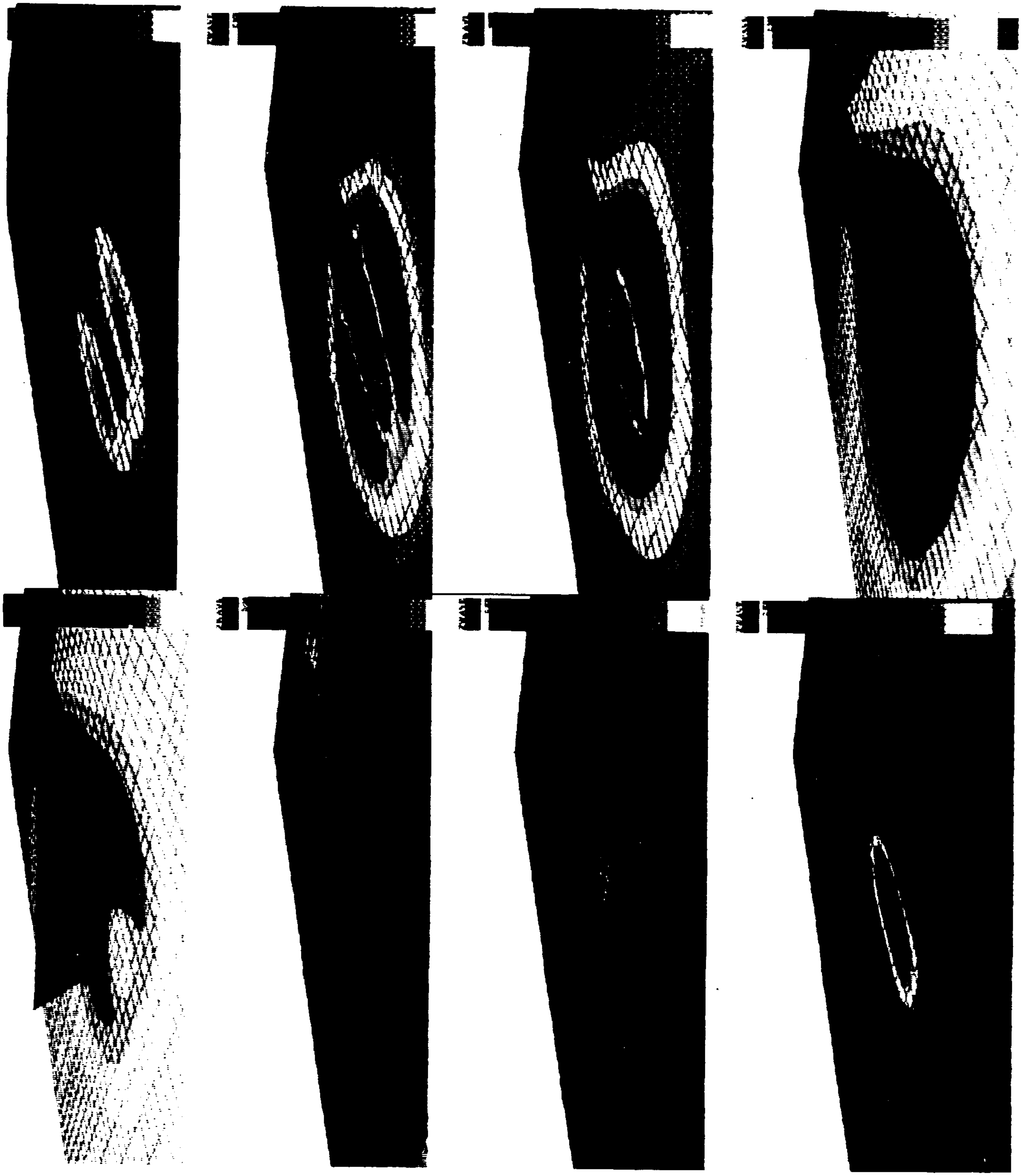


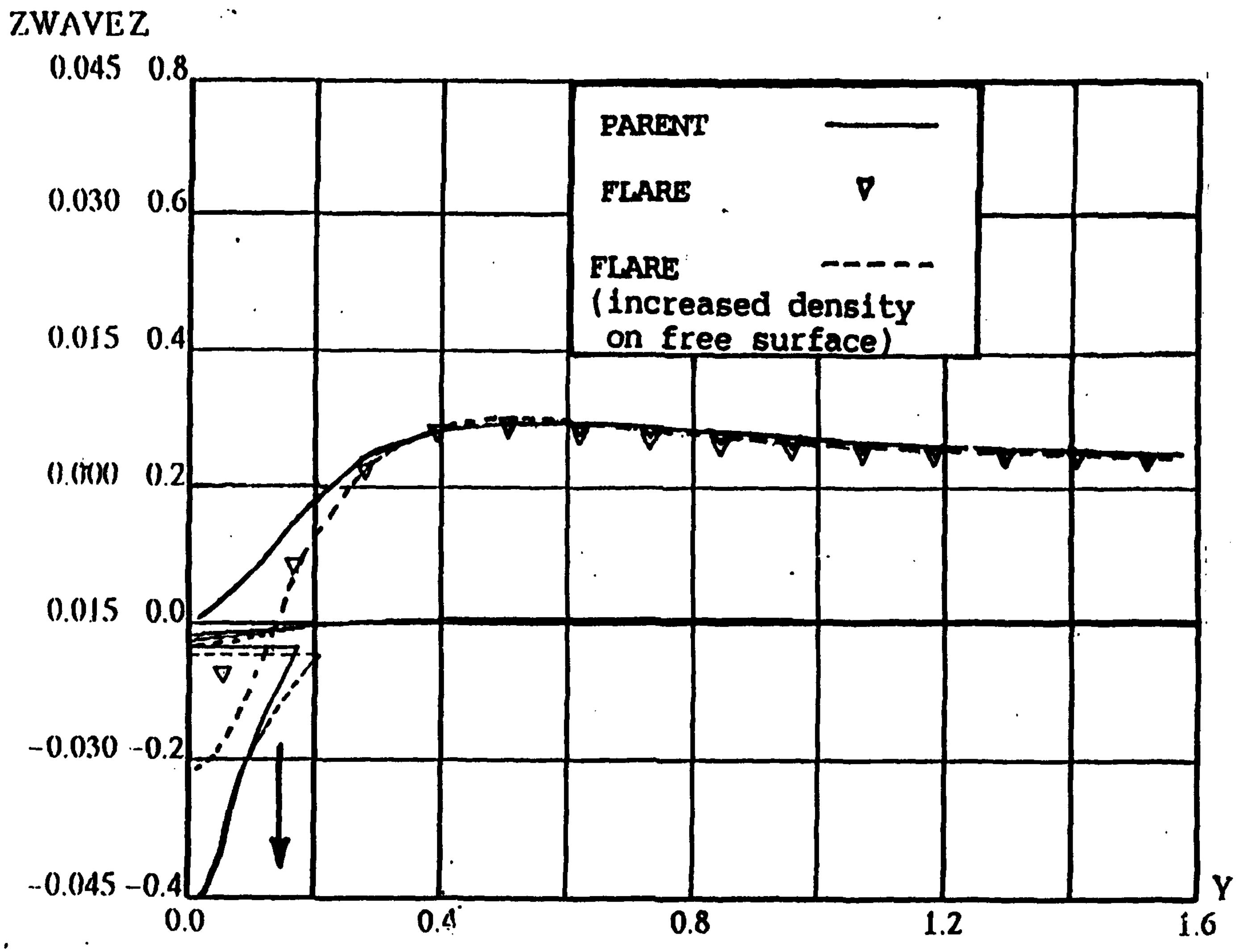
RESPONSE OF PARENT vs FLARED: H/λ= 0.034
PARENT
FLARED λ/L =1.2 Fn=0.3
PITCH (DEG)
6
4
2
0
-2
-4
-6
-8
0
2
4
6
8
10
HEAVE (M)
0.3
0.2
0.1
0.0
-0.1
TIME (seconds)

RESPONSE OF PARENT vs FLARED: $H/\lambda = 0.034$



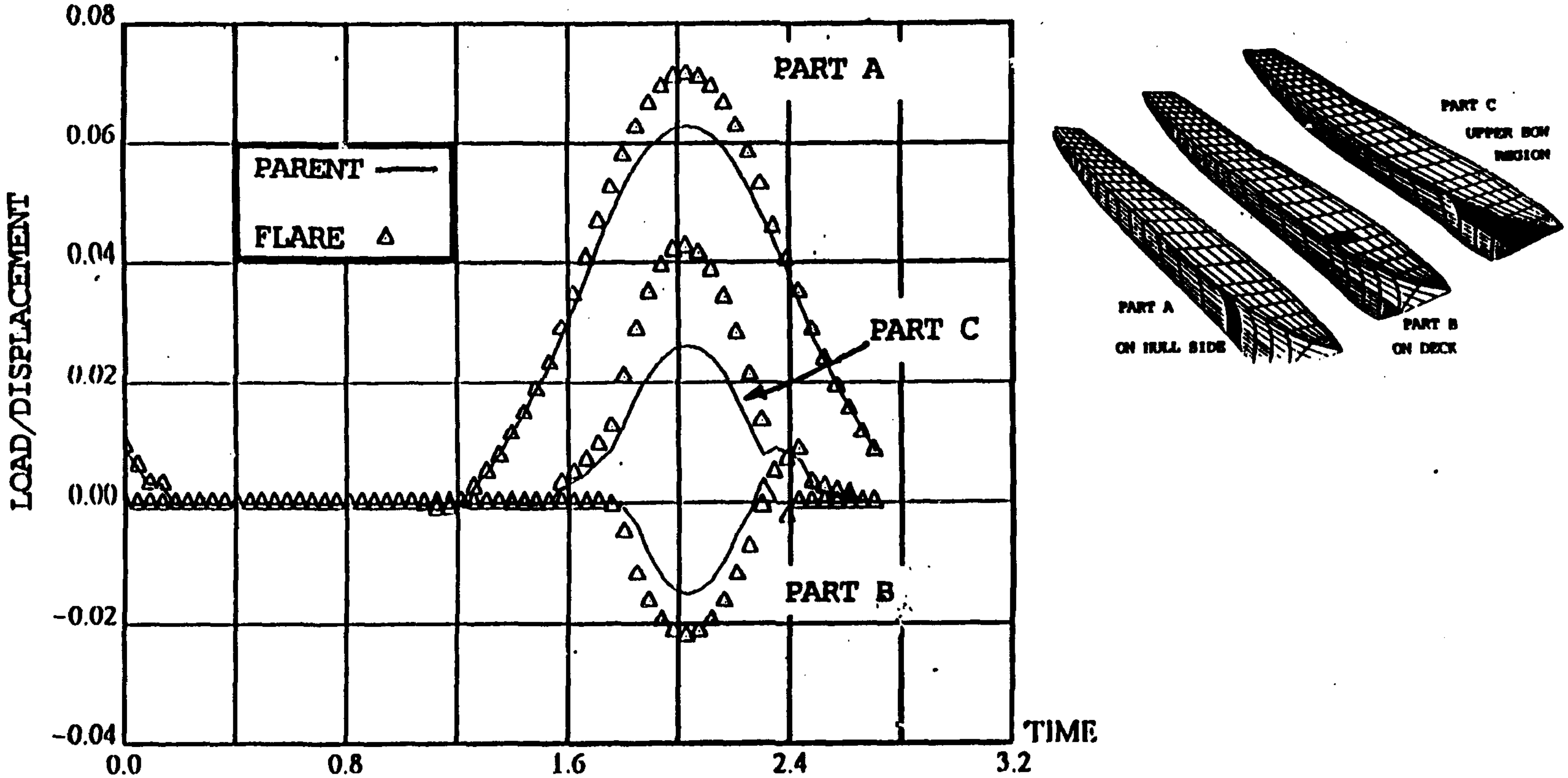
AMI





STATION CUT X= -3.07

AMI



FRIGATE (parent) FR=0.3 p=1.2 l;USAERO/FSP calc.

May 17 16:26:24 1992
OMNI3D (AMI)

AMI

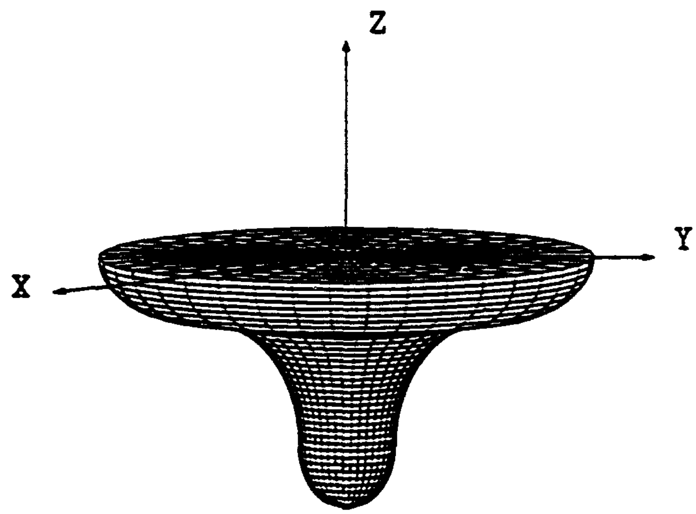
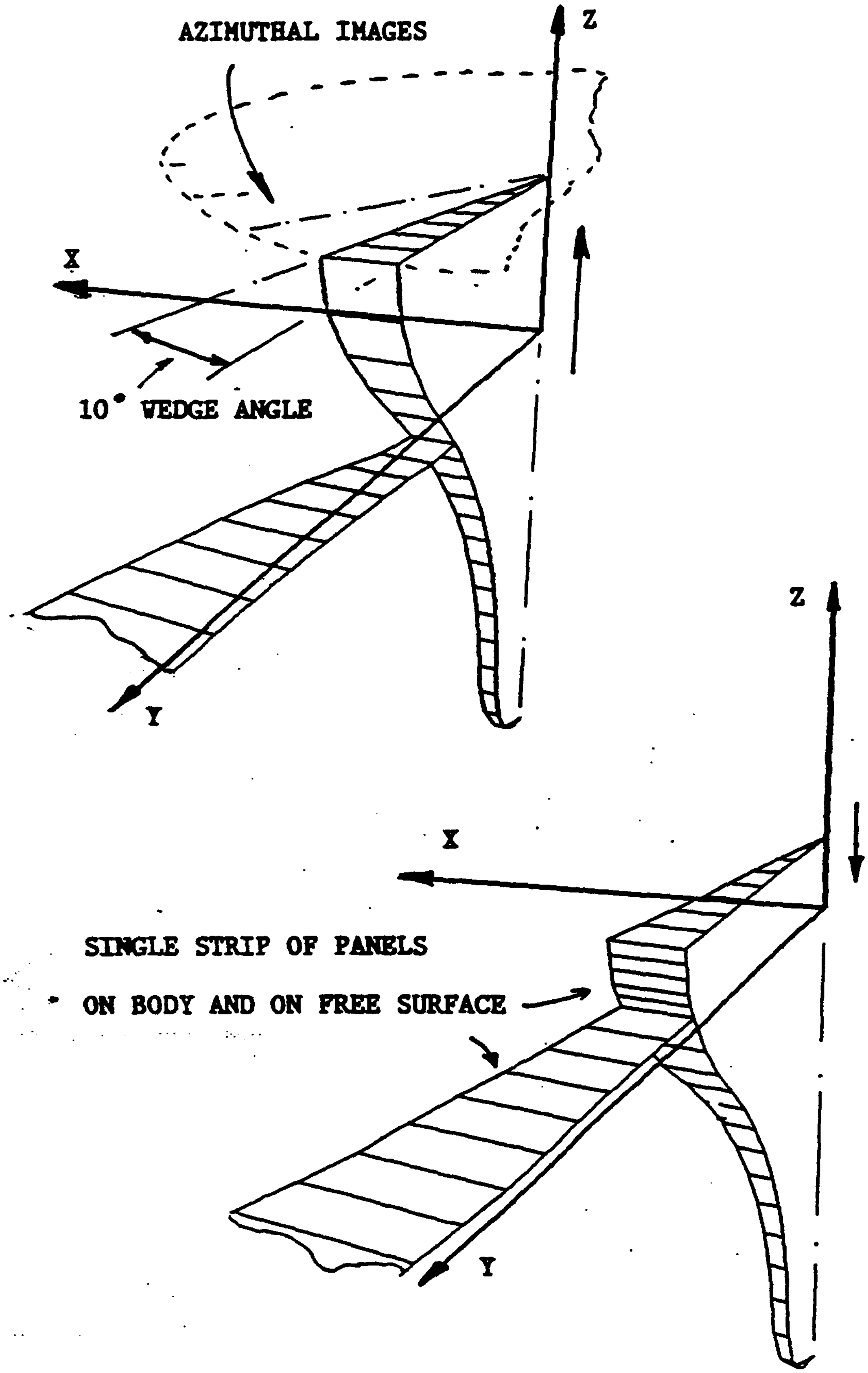


Figure 1: Description of Coordinate Systems and Geometry of the Body

Figure 2: Photograph of the Oscillating Body and its Supporting Structure


AZIMUTHAL IMAGES
Z
X
10° WEDGE ANGLE
Y
Z
X
SINGLE STRIP OF PANELS
ON BODY AND ON FREE SURFACE
Y

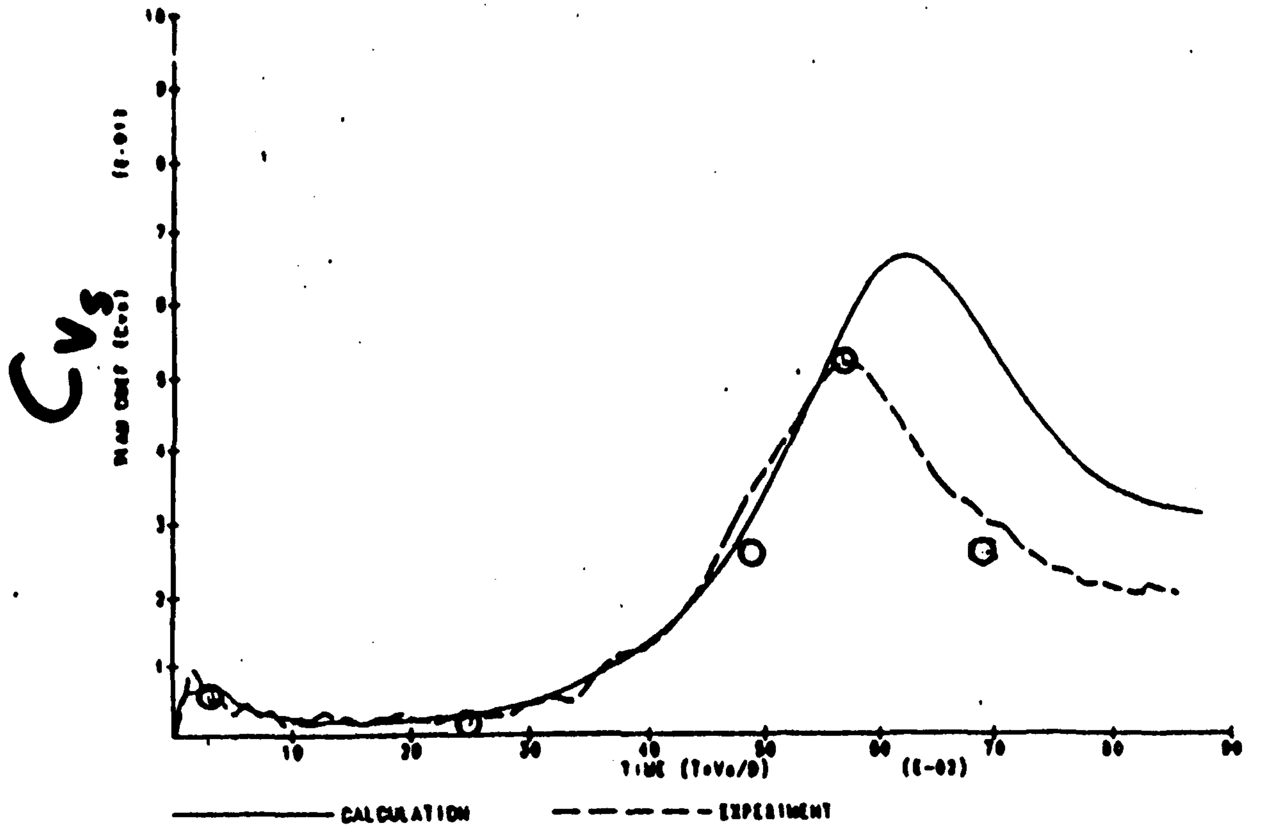
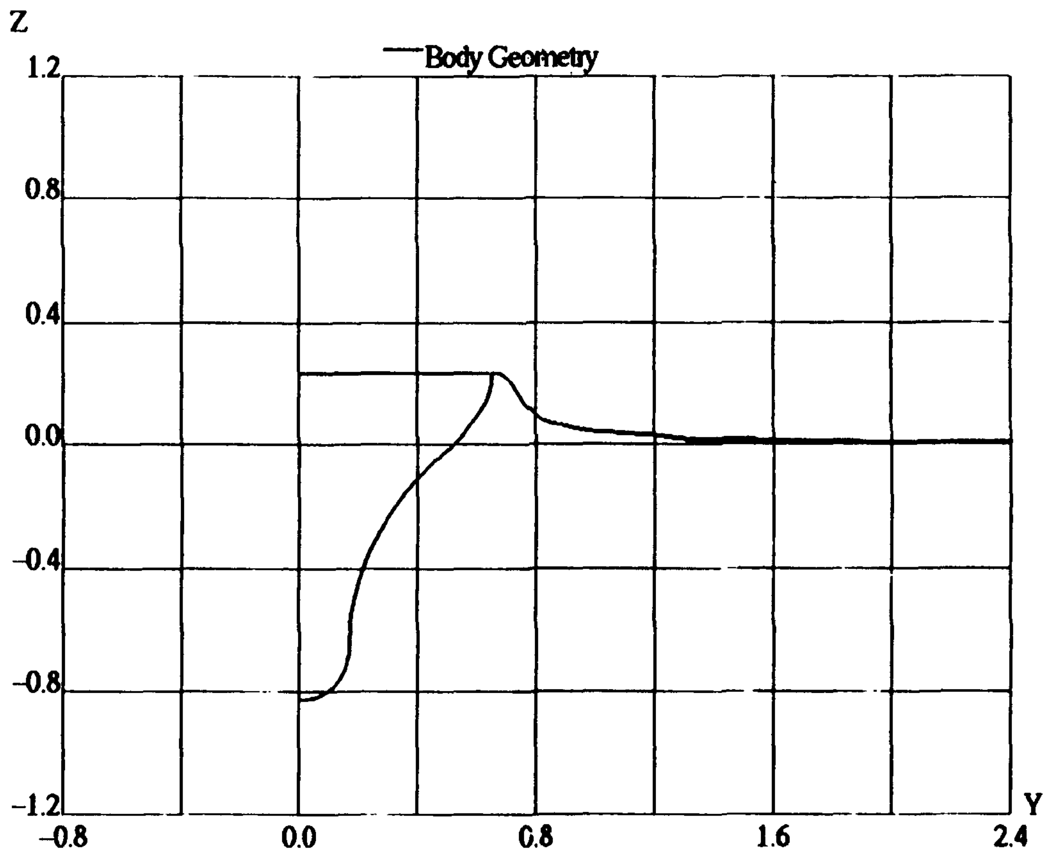


Figure 17b: Comparison Between Theory and Experiment of the Vertical Slam Coefficient for a Cusped Body (FN=3.5371).

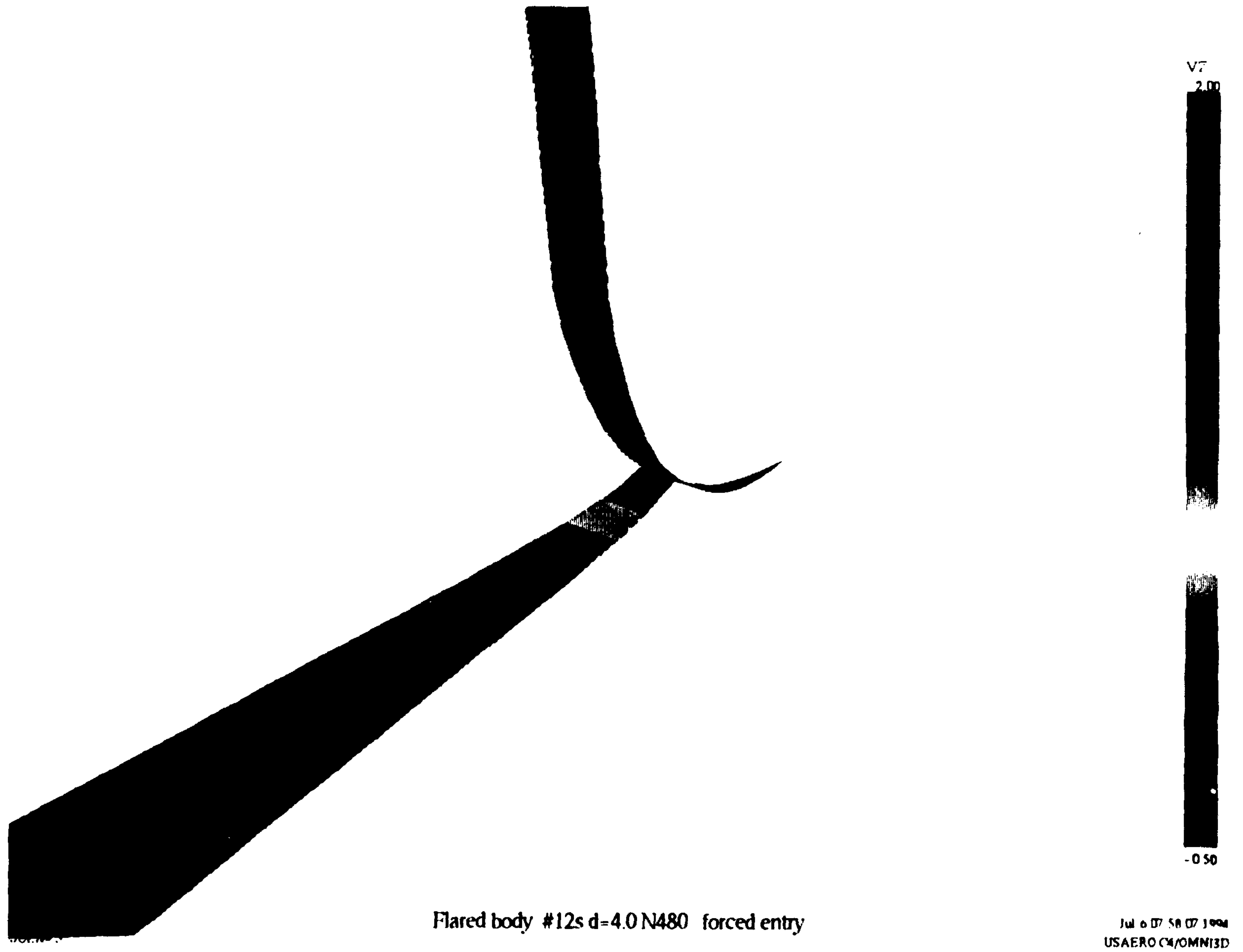


STATION CUT X= 0.00

Flared body: drop height 4 ft

SOLN= 90

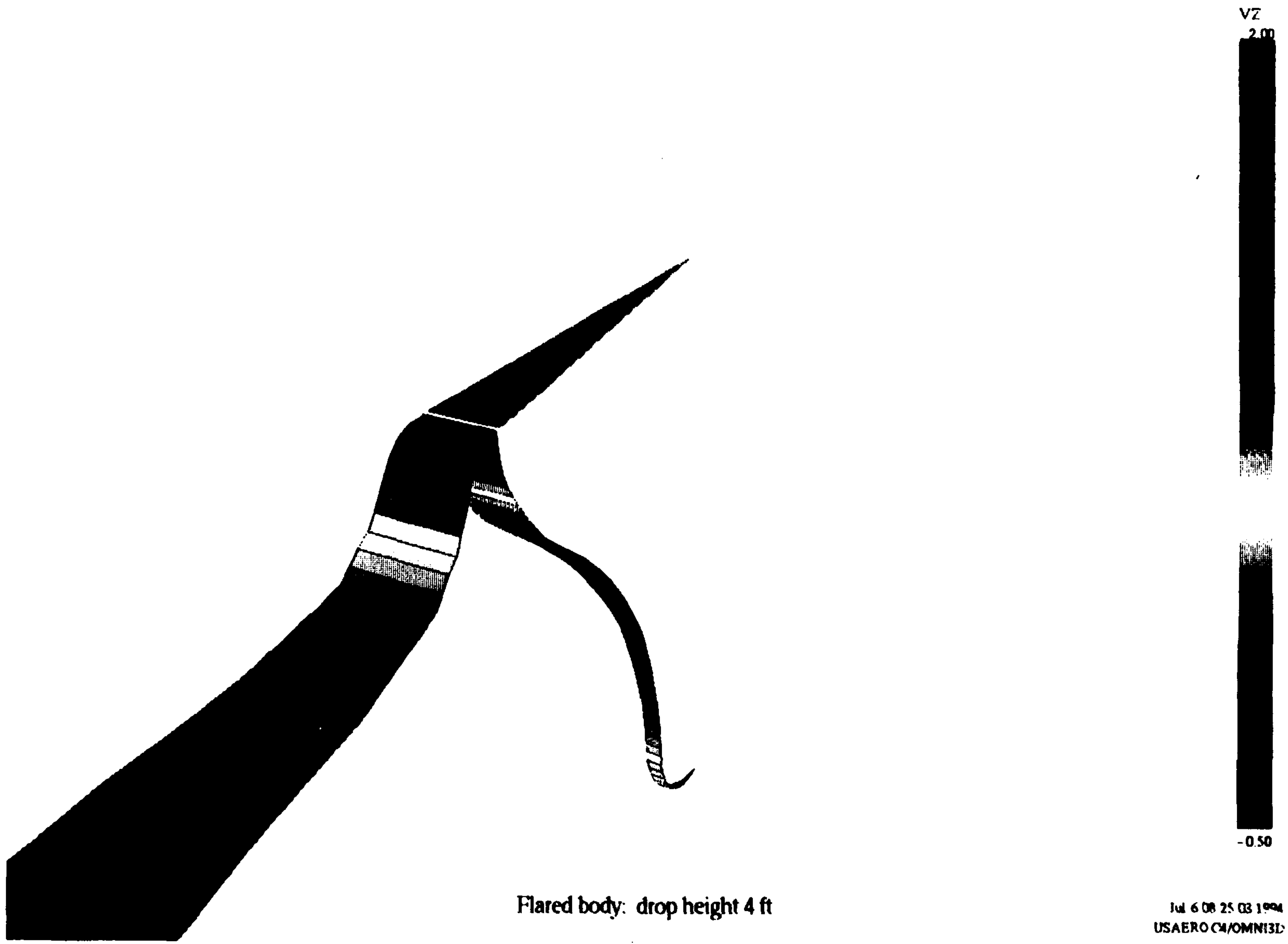
Jul 6 08:35 27 1994
USAERO C4/OMNI3D



Flared body #12s d=4.0 N480 forced entry

Flared body: drop height 4 ft

Flared body: drop height 4 ft



Flared body: drop height 4 ft

UNIVERSITY OF MICHIGAN

FULLY NONLINEAR HYDRODYNAMIC LOADS USING DESINGULARIZED METHODS

Robert F. Beck
Armin W. Troesch
Yusong Cao
Steve Scorpio
Minglun Wang

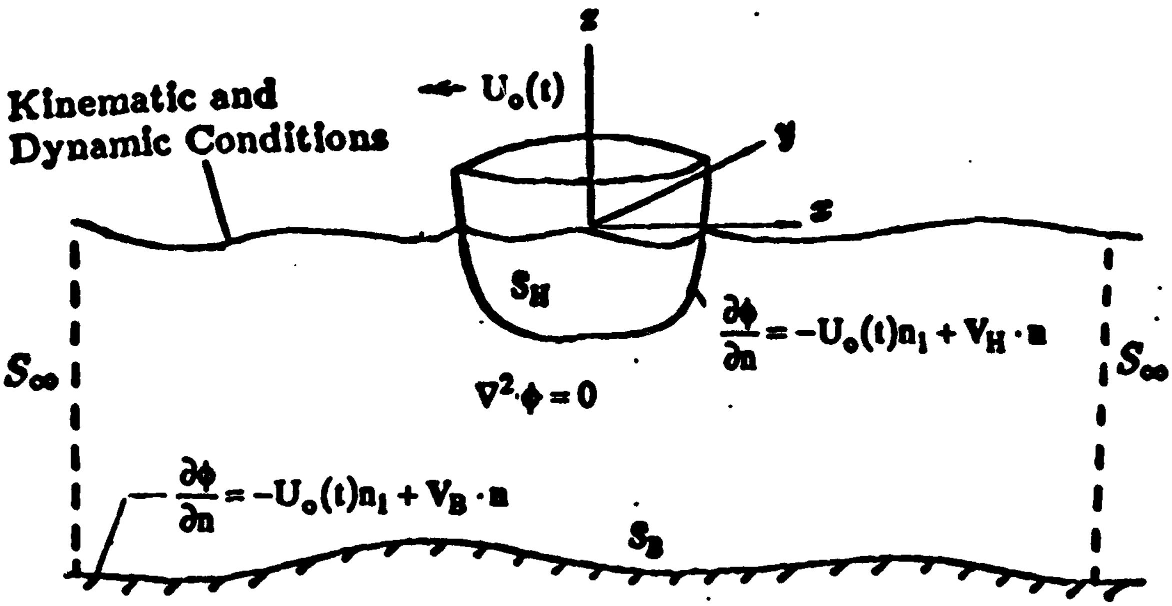
PROBLEM FORMULATION

* Basic Assumptions:

1. **Incompressible and inviscid fluid**
2. **Irrotational flow**
3. **Surface tension neglected**

* Initial boundary value problem:

$$\Phi = U_o(t)x + \phi(x,y,z,t)$$



Initial Conditions:

$$\phi = 0 \qquad (t \leq 0)$$

$$\eta = 0 \qquad (t \leq 0)$$

FREE SURFACE BOUNDARY CONDITIONS

- **Kinematic condition:**

$$\frac{\delta \eta}{\delta t} = \frac{\partial \phi}{\partial z} - (\nabla\phi - \mathbf{v}) \cdot \nabla\eta - U_o(t)\frac{\partial \eta}{\partial x} \qquad \text{(on F.S.)}$$

$$\frac{\delta x}{\delta t} = v_x \qquad \text{(on F.S.)}$$

$$\frac{\delta y}{\delta t} = v_y \qquad \text{(on F.S.)}$$

- **Dynamic condition:**

$$\frac{\delta \phi}{\delta t} = -g\eta - \frac{1}{2}\nabla\phi \cdot \nabla\phi + \mathbf{v} \cdot \nabla\phi - \frac{P_a}{\rho} - U_o(t)\frac{\partial \phi}{\partial x} \qquad \text{(on F.S.)}$$

where $\frac{\delta}{\delta t} = \frac{\partial}{\partial t} + \mathbf{v} \cdot \nabla$ is the time derivative following a node moving with velocity **v** .

- **Fixed Horizontal Nodes** $\left(\mathbf{v}=\left(0\ \ 0,\ \frac{\partial\eta}{\partial t}\right)\right)$

$$\frac{\partial\eta}{\partial t}=\frac{\partial\phi}{\partial z}-\nabla\phi\cdot\nabla\eta-U_o(t)\frac{\partial\eta}{\partial x} \qquad \text{(or F.S.)}$$

and

$$\frac{\delta\phi}{\delta t}=-g\eta-\frac{1}{2}\nabla\phi\cdot\nabla\phi+\frac{\partial\eta}{\partial t}\frac{\partial\phi}{\partial z}-\frac{P_a}{\rho}-U_o(t)\frac{\partial\phi}{\partial x}$$

- **Material Nodes** $(\mathbf{v}=U_o(t)\mathbf{i}+\nabla\phi)$

$$\frac{D\mathbf{X}_F}{Dt}=U_o(t)\mathbf{i}+\nabla\phi$$

and

$$\frac{D\phi}{Dt}=-g\eta+\frac{1}{2}\nabla\phi\cdot\nabla\phi-\frac{P_a}{\rho}$$

where $\mathbf{x}_F(t)=(x_F(t),y_F(t),z_F(t))$ is the position vector of a fluid particle on F.S. and

$$\frac{D}{Dt}=\frac{\partial}{\partial t}+\nabla\Phi\cdot\nabla$$

is the material derivative.

FULLY NONLINEAR SOLUTION METHOD

Time-Stepping Procedure

1. Solve a mixed BVP at a given instant of time by a desingularized boundary integral method.

2. Integrate the nonlinear free surface kinematic and dynamic conditions with respect to time.

3. For free body problem, must compute $\frac{\delta\phi}{\delta t}$ or $\frac{\partial\phi}{\partial t}$ at present time step, then integrate the body equations of motion.

DESINGULARIZED BOUNDARY INTEGRAL METHOD

- Simple sources can be used because of desingularization
- Easy discretization (nodes only, no panels)
- Desingularization distance related to the local node spacing
- Simple algorithm, easy programming and high performance on supercomputers
- Can lead to O(N) algorithm

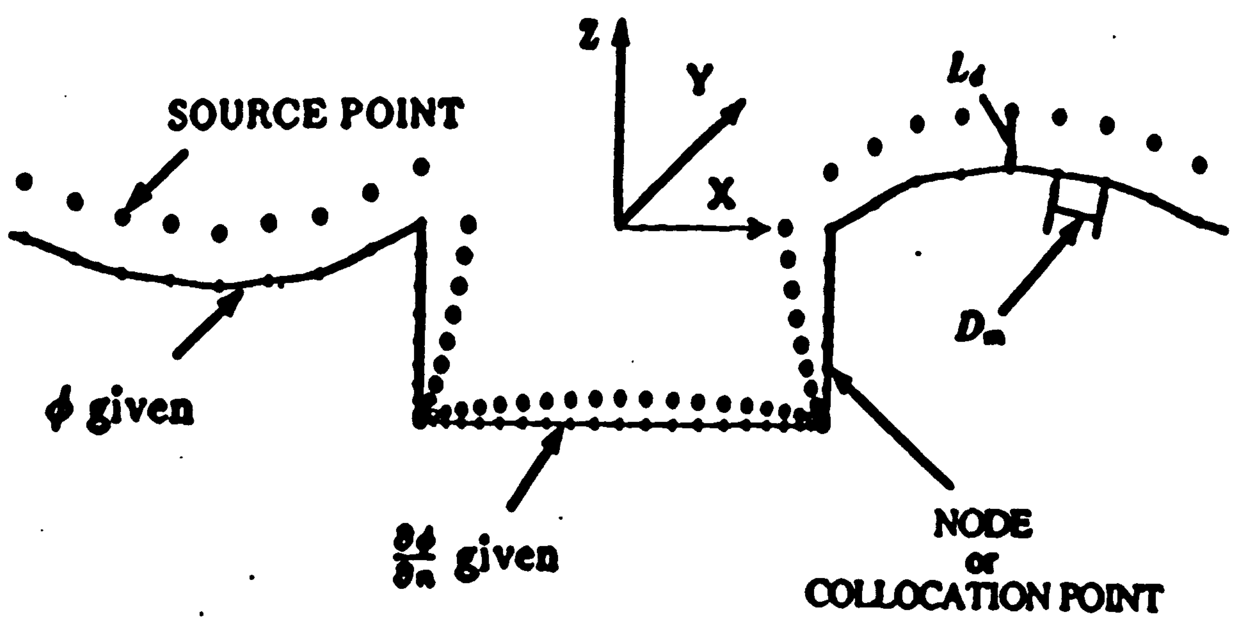


Figure 1: Schematic of source and node locations

HYDRODYNAMIC FORCES

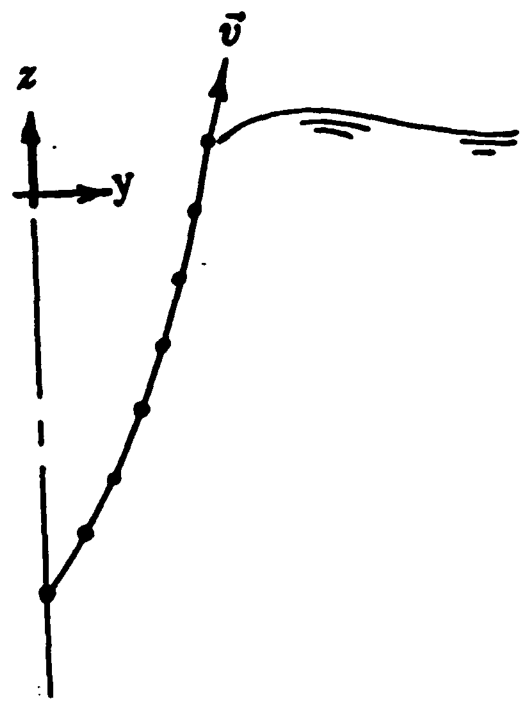
$$F_i = -\iint_s p\, n_i\, ds$$

where

$$\frac{P}{\rho} = -\frac{\partial \phi}{\partial t} - U_o(t)\frac{\partial \phi}{\partial x} - gz - \frac{1}{2}\nabla\phi \cdot \nabla\phi$$

$$= -\frac{\delta \phi}{\delta t} - U_o\frac{\partial \phi}{\partial x} - gz - \frac{1}{2}\nabla\phi \cdot \nabla\phi + v \cdot \nabla\phi$$

ϕ = perturbation potential in moving coordinate system.



APPLICATIONS OF THE FULLY NONLINEAR DESINGULARIZED METHOD

- **Verification by comparison with analytic solutions for flows generated by isolated singularities and bodies of simple geometry (Cao, Schultz, and Beck 1991 and Cao, Lee, and Beck 1992)**

- **Shallow water solitons due to a moving disturbance (Cao, Beck, and Schultz 1993a)**

- **Two-dimensional wave tank with an adsorbing beach (Beck, Cao, and Lee 1993 and Cao, Beck, and Schultz 1993b)**

- **Submerged spheroid traveling at constant forward speed (Bertram, Schultz, Cao, and Beck 1991)**

- **Two-dimensional added mass and damping for forced heave, sway and roll (Lee 1992, and Beck, Cao, and Lee 1993)**

- **Two-dimensional heave, sway and roll motions of a rectangular body due to incident waves (Cao, Beck, and Schultz 1994)**

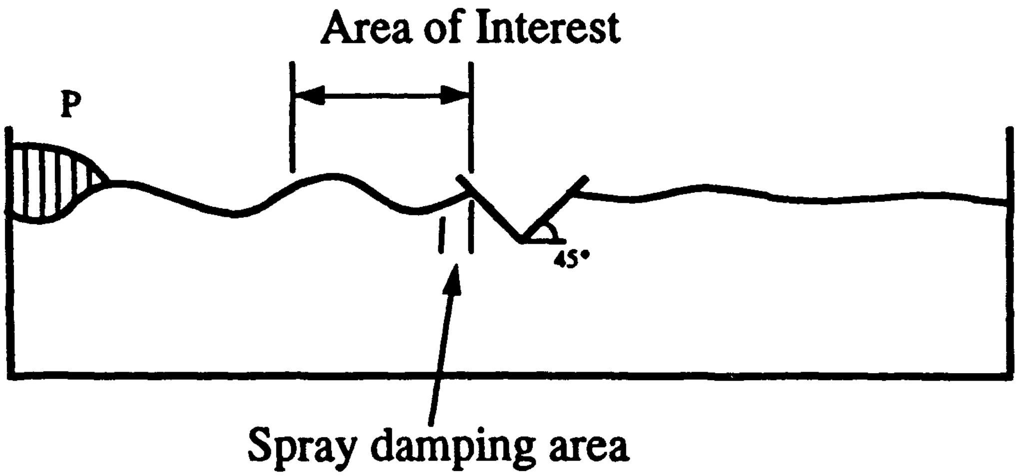
- **Three-dimensional cylinder in forced heave (Beck, Cao, and Lee 1993)**

- **Exciting forces on a Tension Leg Platform**

- **Calm water resistance and added mass and damping of a Wigley hull (Beck, Cao, and Lee 1993, and Beck, Cao, and Scorpio 1994)**

Free Surface Elevations for a 2-d tank with a 45 degree wedge shaped body

with and without spray damping



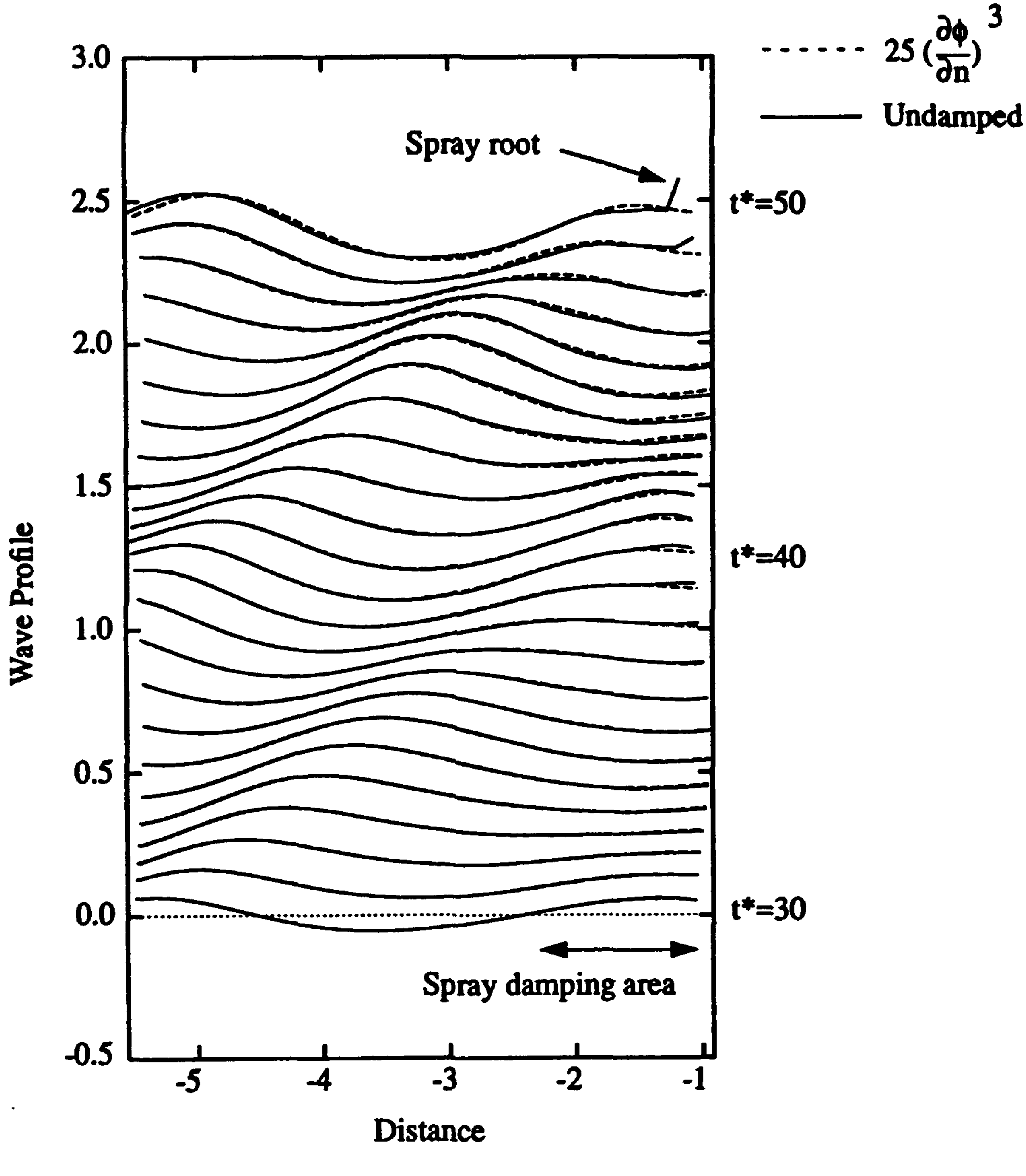
$$L = 30$$

$$H = 3$$

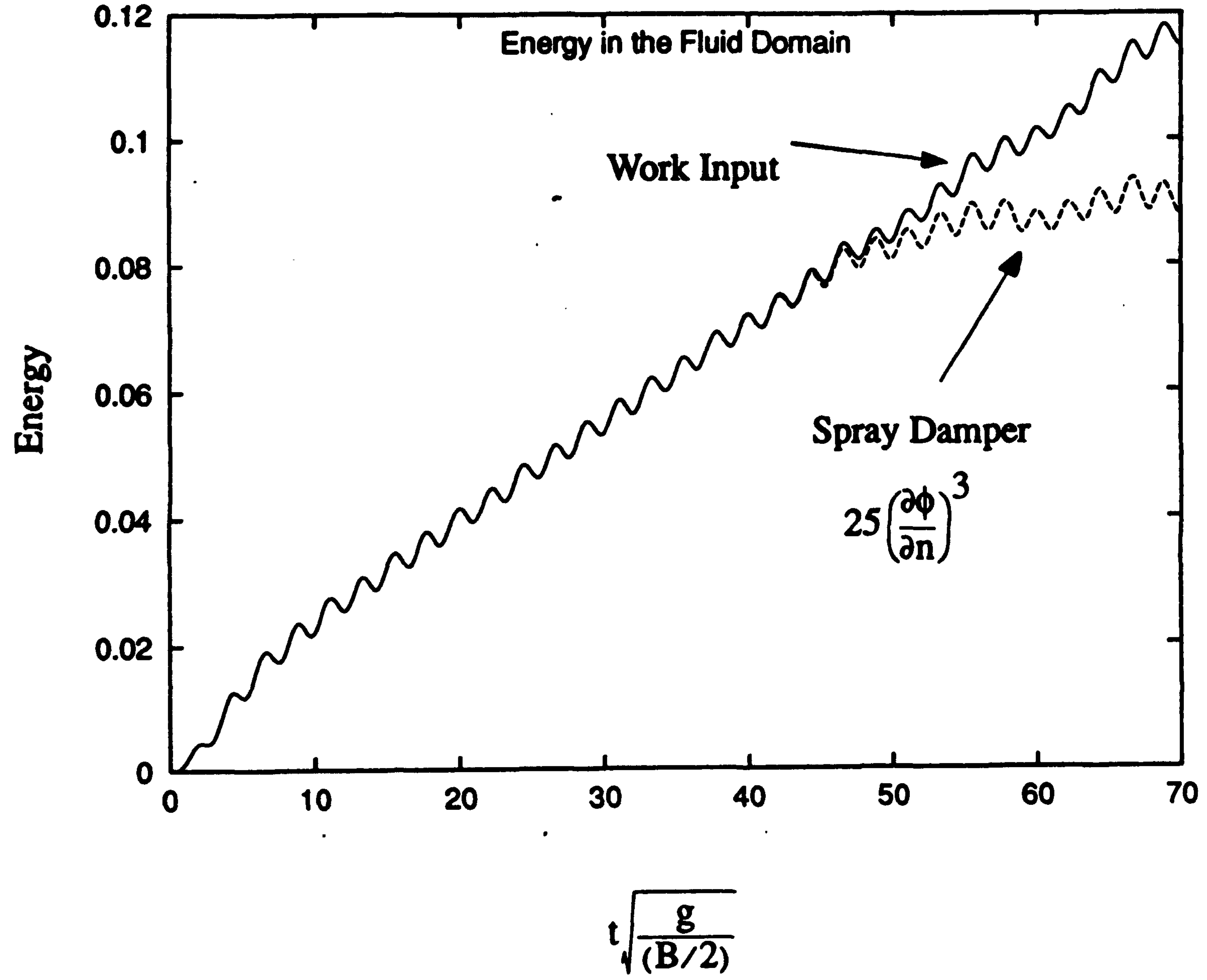
$$B/2 = 1$$

$$\frac{|P|}{B/2} = 0.1$$

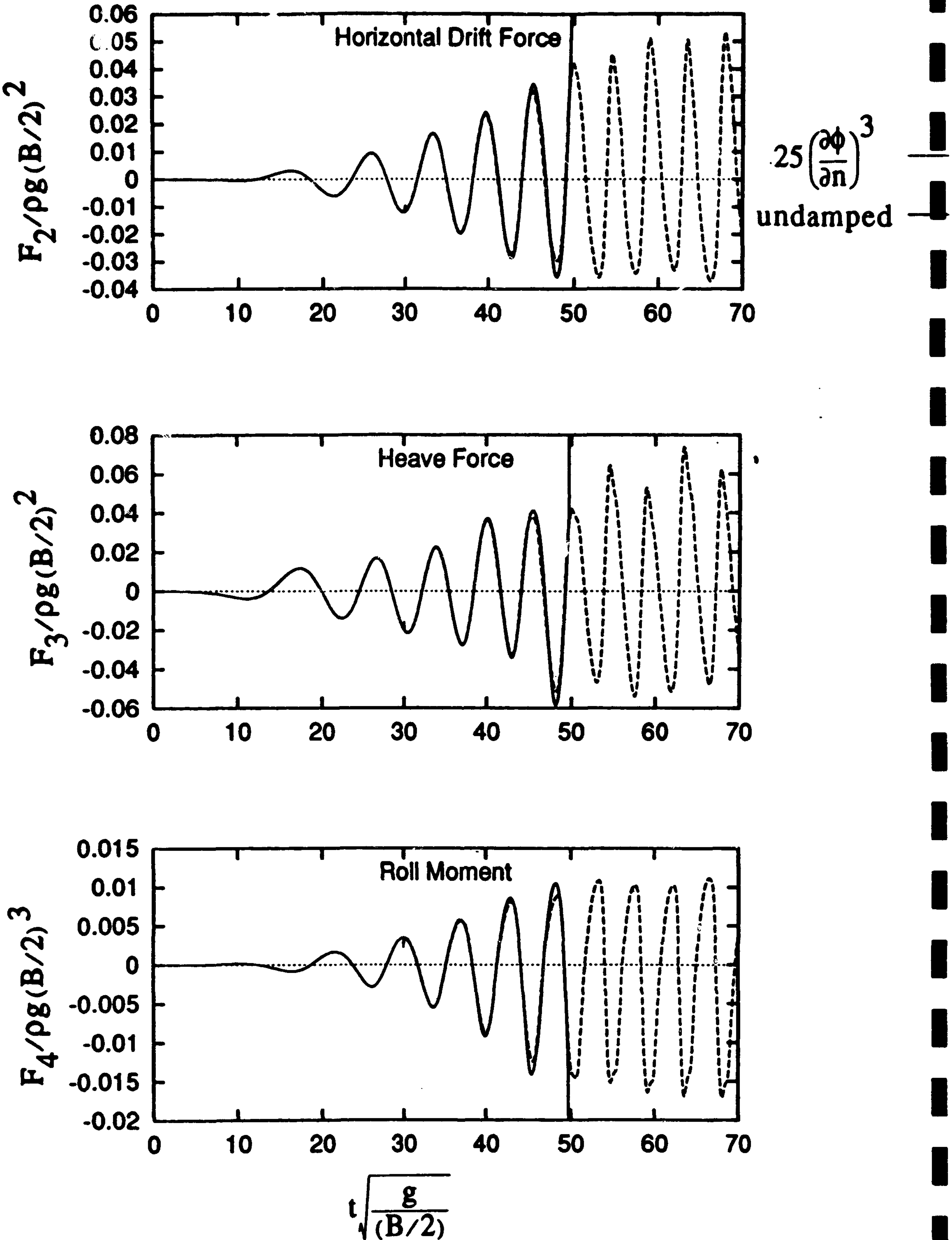
$$\omega\sqrt{\frac{(B/2)}{g}} = \sqrt{2}$$



Wave profiles near the wedge



Energy in the Fluid Domain
Work Input
Spray Damper
$25\left(\frac{\partial\phi}{\partial n}\right)^3$
Energy
0.12
0.1
0.08
0.06
0.04
0.02
0
0
10
20
30
40
50
60
70
$t\sqrt{\frac{g}{(B/2)}}$



Horizontal Drift Force
$F_2/\rho g(B/2)^2$
$.25\left(\frac{\partial\phi}{\partial n}\right)^3$
undamped
Heave Force
$F_3/\rho g(B/2)^2$
Roll Moment
$F_4/\rho g(B/2)^3$
$t\sqrt{\frac{g}{(B/2)}}$

Freely Floating Body Dynamics

Hydrodynamic problem

$$\begin{cases} \dfrac{dX_F}{dt} = F_1(X_F, \phi_F, X_G, V_G) \\ \dfrac{d\phi_F}{dt} = F_2(X_F, \phi_F, X_G, V_G) \end{cases}$$

Euler's equation of motion

$$\begin{cases} \dfrac{dX_G}{dt} = V_G \\ \dfrac{dV_G}{dt} = F_3\left(X_F, \phi_F, X_G, V_G, \dfrac{dV_G}{dt}\right) \end{cases}$$

where the state variables are defined by the generalized vectors:

X_F = location of free surface nodes
ϕ_F = potential of free surface nodes
X_G = location of vessel center of gravity in 6-degrees of freedom
V_G = 6 components of body velocity at center of gravity

Hydrodynamic force at present time step

$$F_i = -\iint_s p\, n_i\, ds$$

$$\frac{p}{\rho} = -\frac{\partial\phi}{\partial t} - U_o(t)\frac{\partial\phi}{\partial x} - gz - \frac{1}{2}\nabla\phi\cdot\nabla\phi$$

$$= -\frac{\delta\phi}{\delta t} - U_o(t)\frac{\partial\phi}{\partial x} - gz - \frac{1}{2}\nabla\phi\cdot\nabla\phi + \mathbf{v}\cdot\nabla\phi$$

To evaluate $\frac{\partial\phi}{\partial t}$ or $\frac{\delta\phi}{\delta t}$

- Backwards differencing for $\frac{\delta\phi}{\delta t}$
- Direct solution of $\frac{\partial\phi}{\partial t}$ BVP

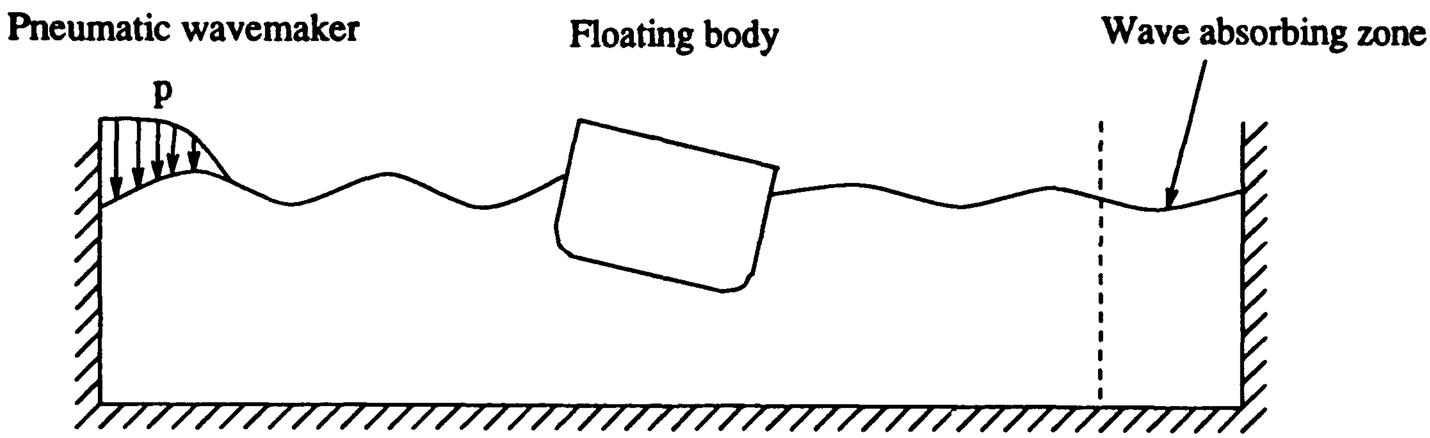
$$\nabla^2\left(\frac{\partial\phi}{\partial t}\right) = 0 \quad \text{in fluid}$$

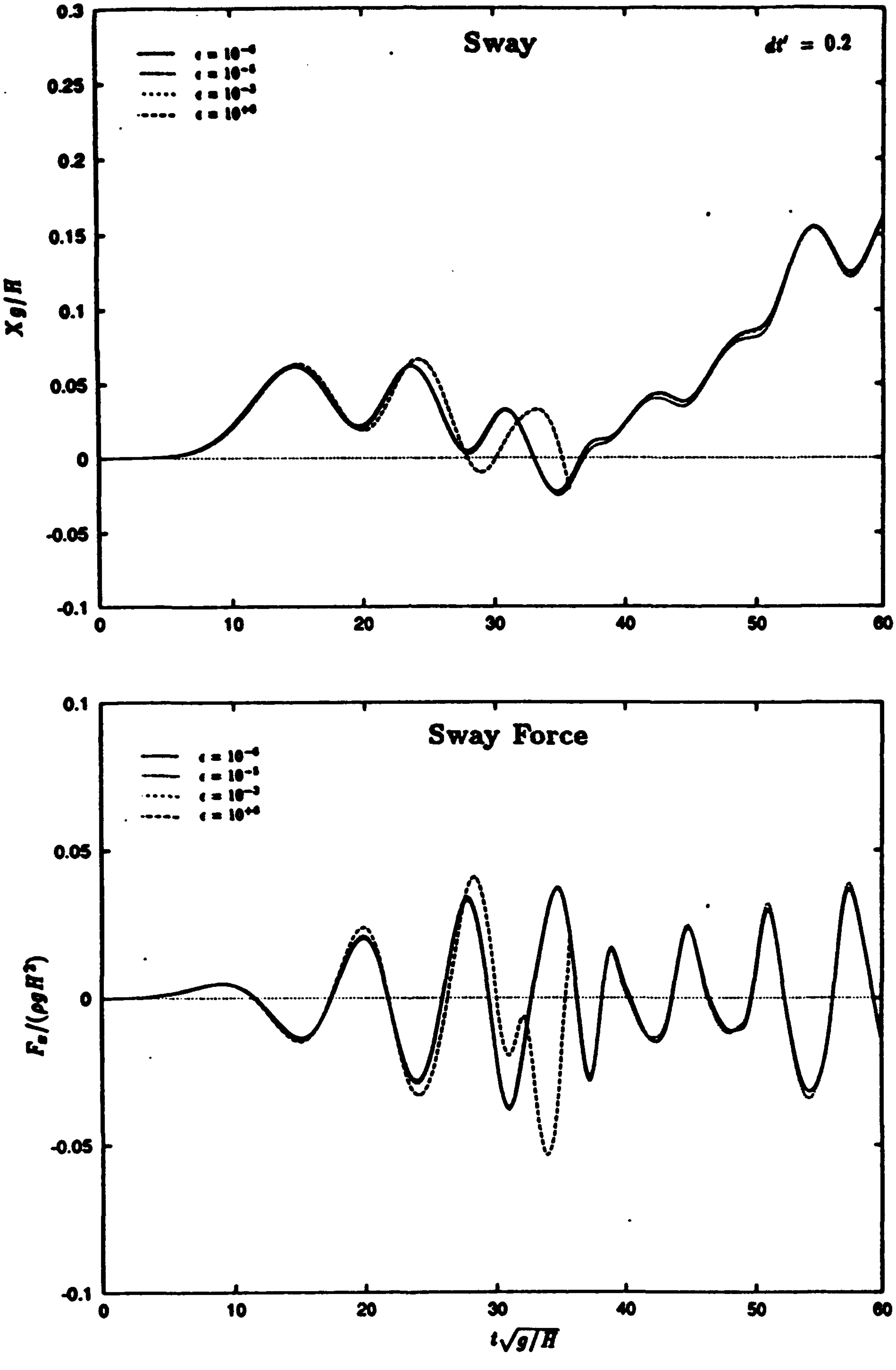
$$\frac{\partial\phi}{\partial t} = -g\eta - U_o(t)\frac{\partial\phi}{\partial x} - \frac{1}{2}\nabla\phi\cdot\nabla\phi - \frac{P_a}{\rho} \quad \text{on } S_F$$

$$\frac{\partial}{\partial n}\left(\frac{\partial\phi}{\partial t}\right) = \frac{\partial \mathbf{n}}{\partial t}\cdot(\mathbf{v}_H - \nabla\phi) + \mathbf{n}\cdot\frac{\partial \mathbf{v}_H}{\partial t} \quad \text{on } S_H, S_B$$

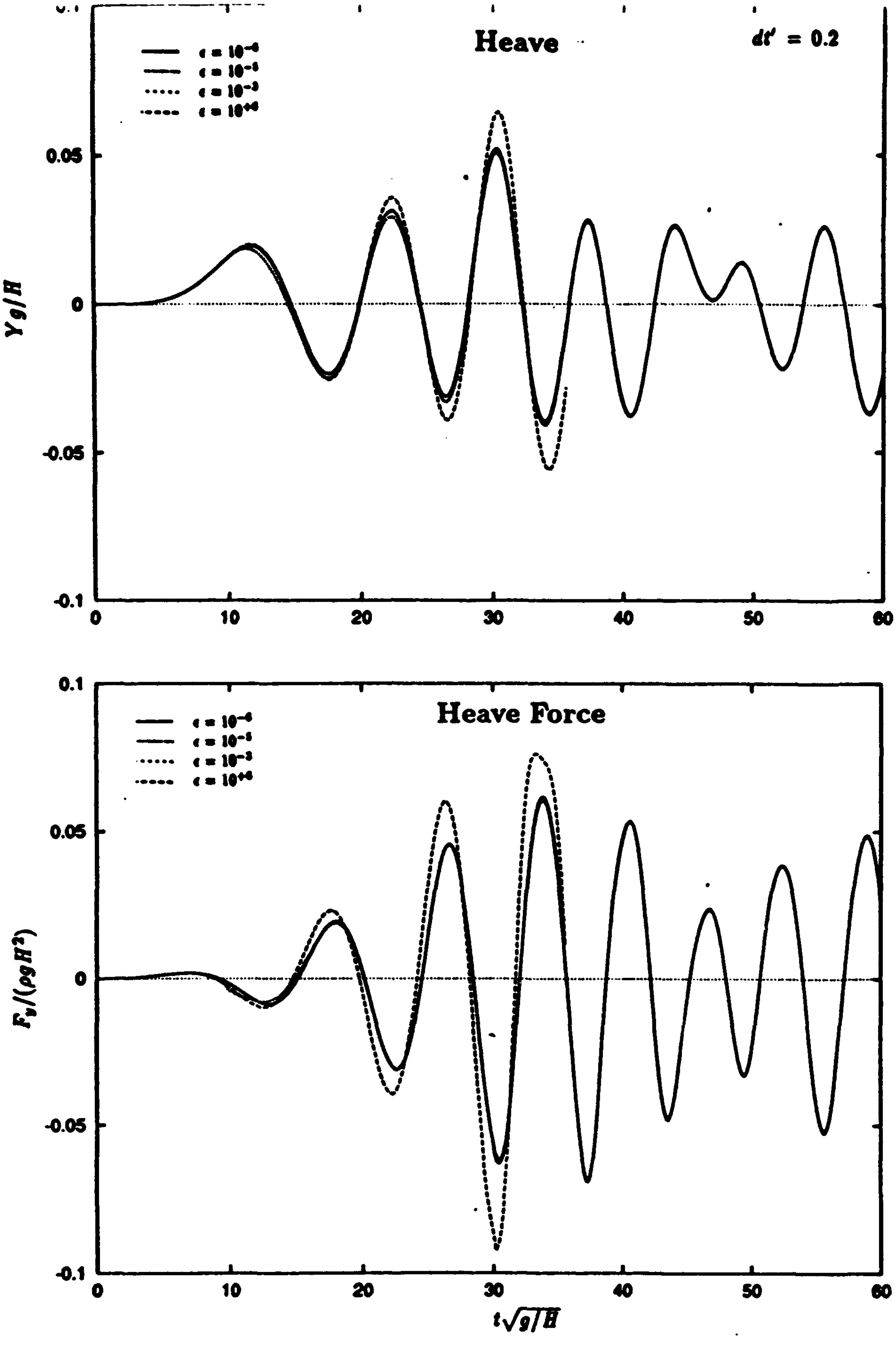
$$\nabla\left(\frac{\partial\phi}{\partial t}\right) \to 0 \qquad R \to \infty$$

WAVE-INDUCED FREE BODY MOTIONS IN A TWO-DIMENSIONAL WAVE TANK

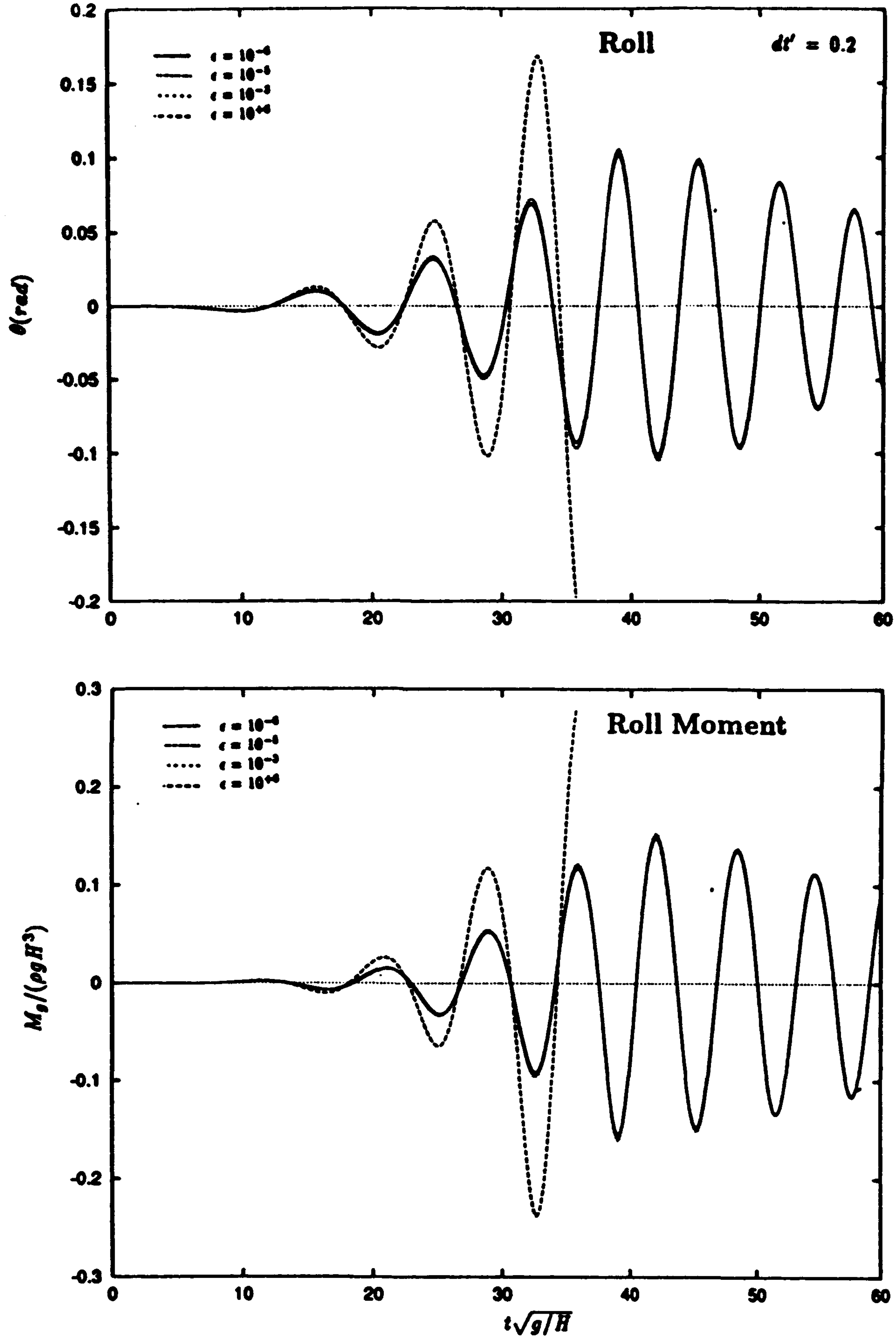




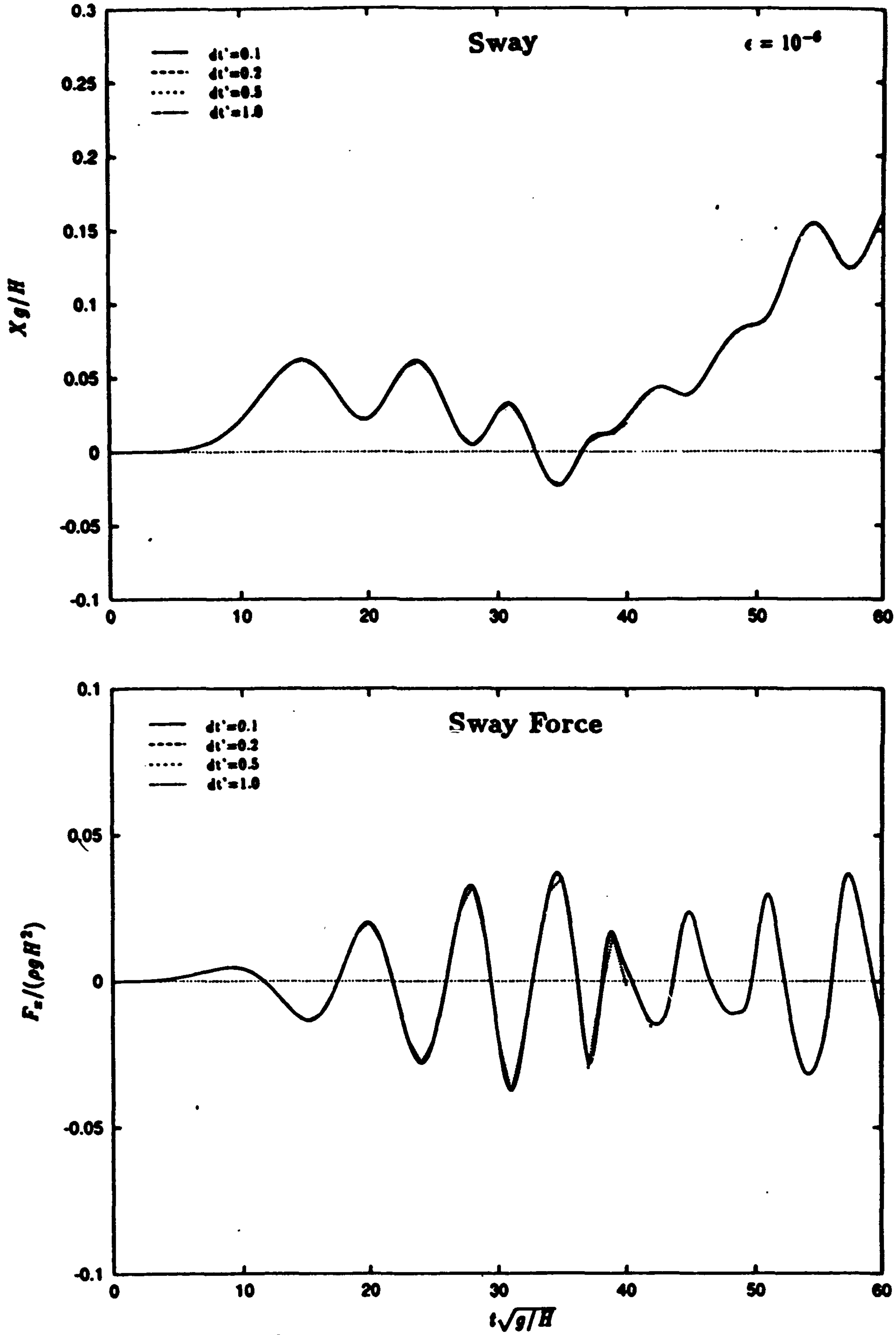
Effect of error tolerance on sway (dt* = 0.2)



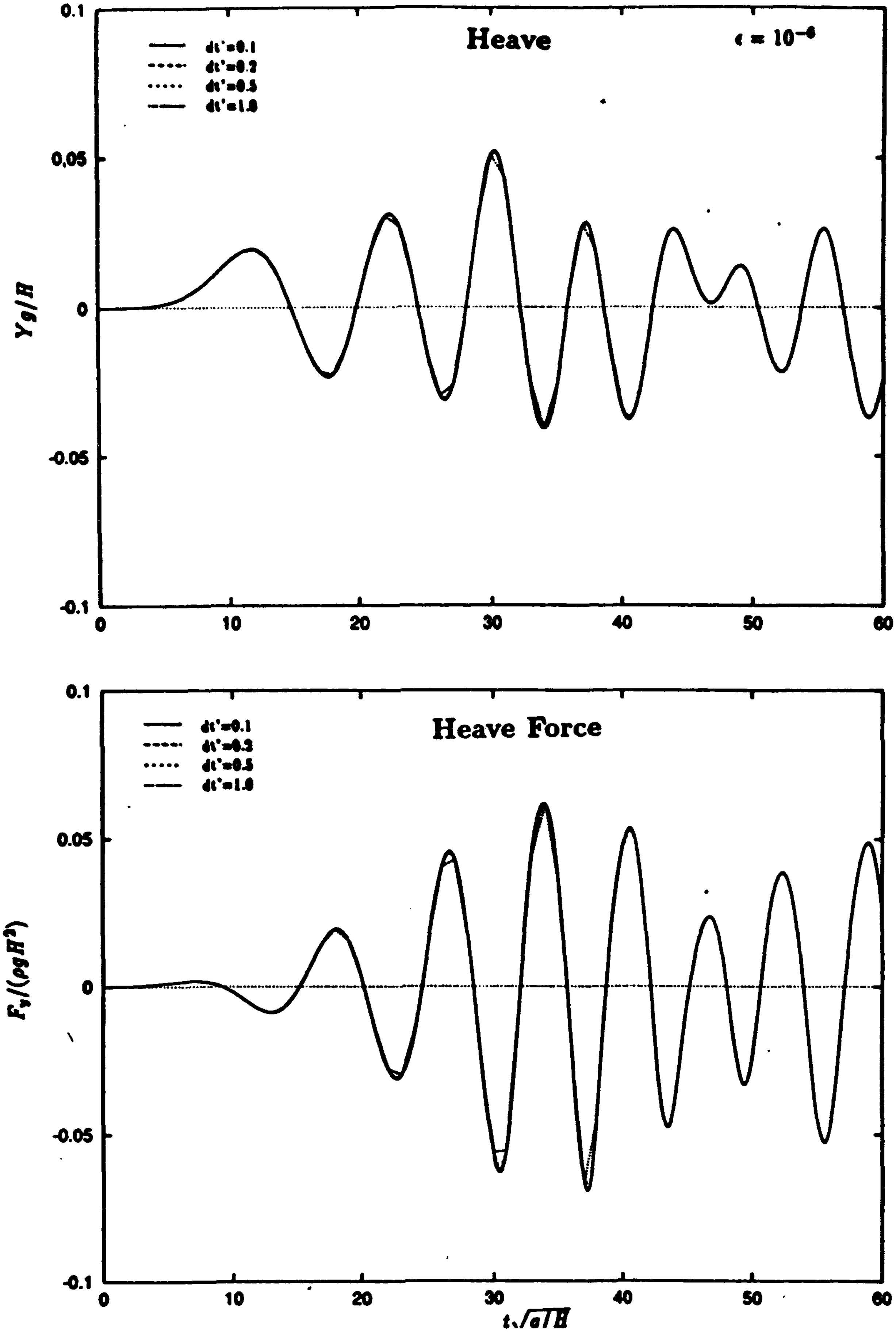
Effect of error tolerance on heave ($dt^* = 0.2$)



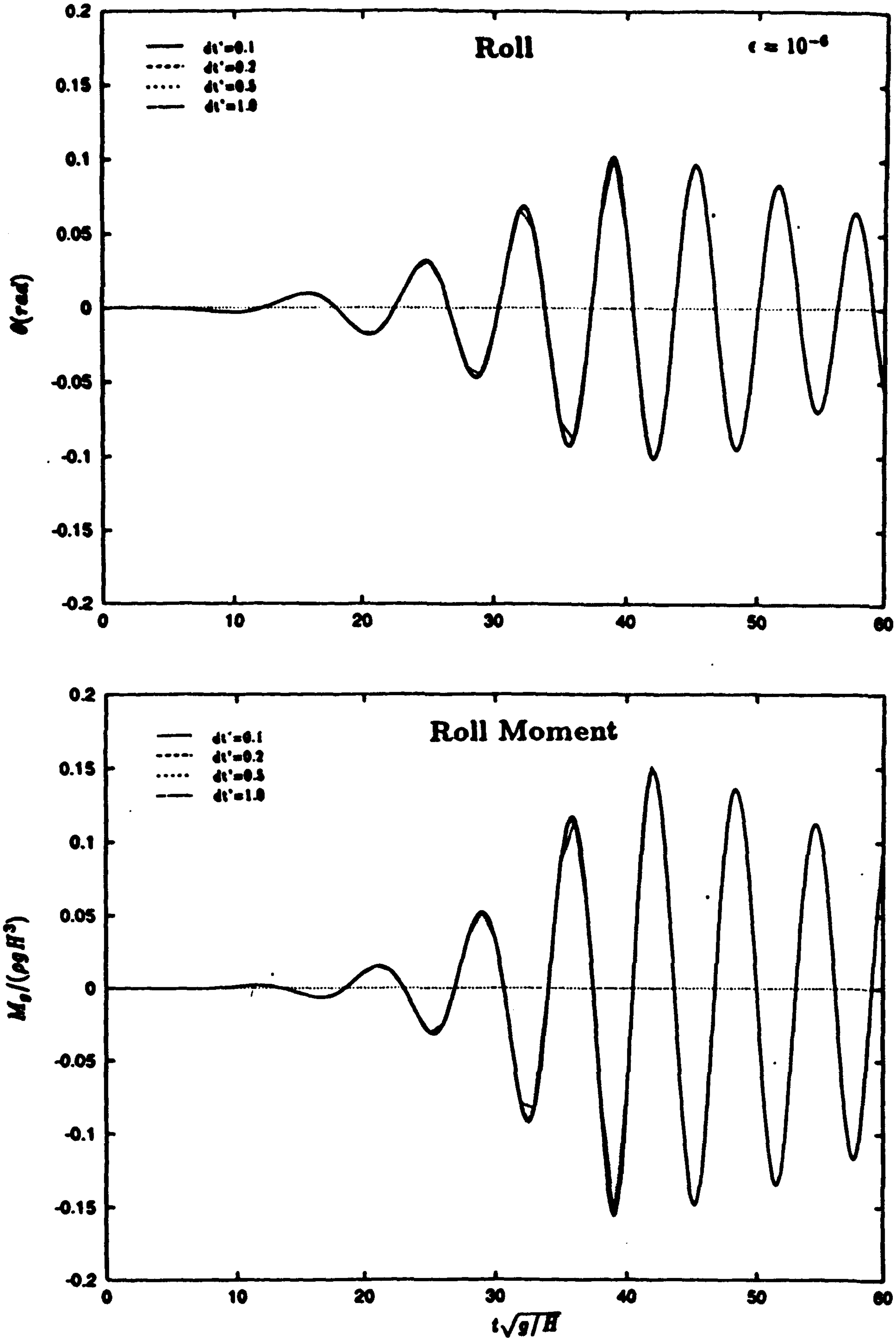
Effect of error tolerance on roll (dt* = 0.2)



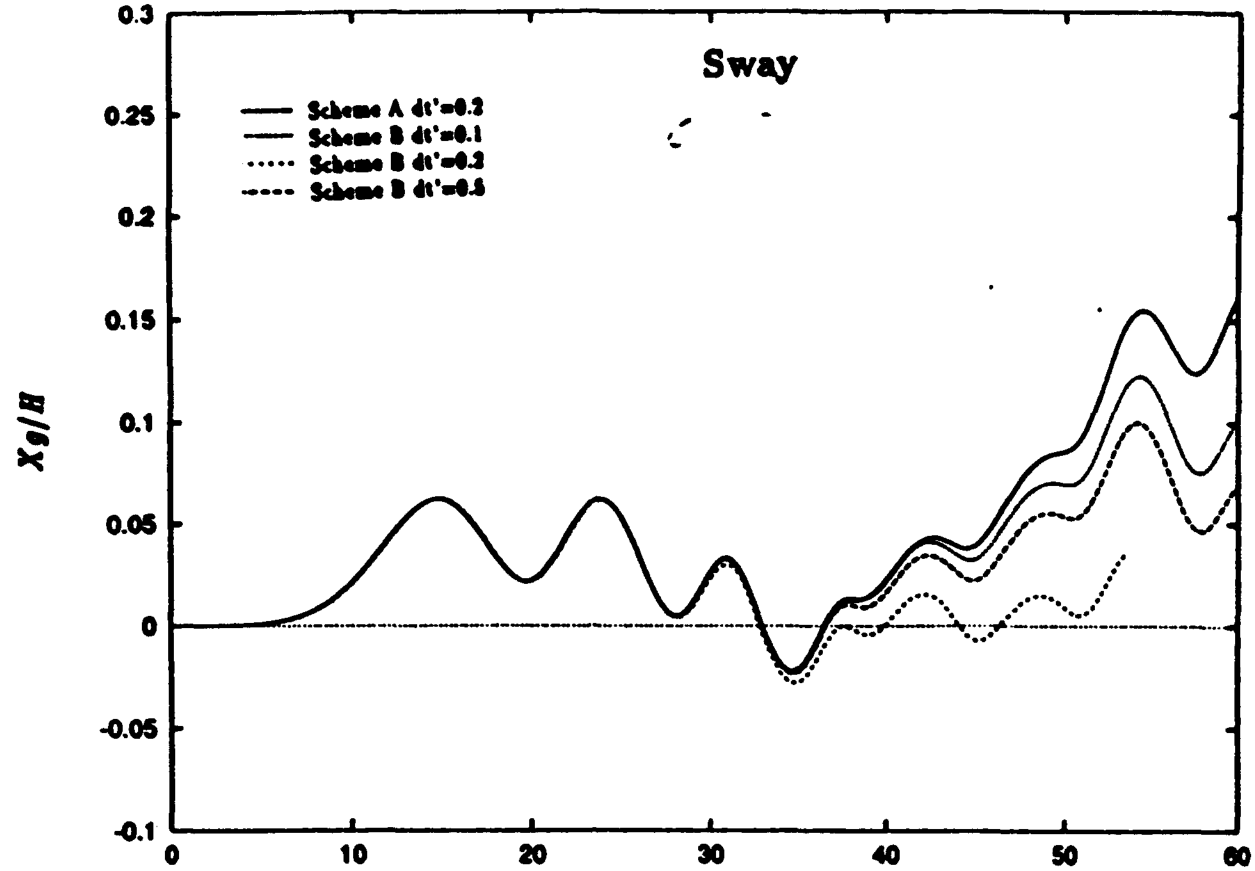
Effect of time step size on sway $(\varepsilon = 10^{-6})$

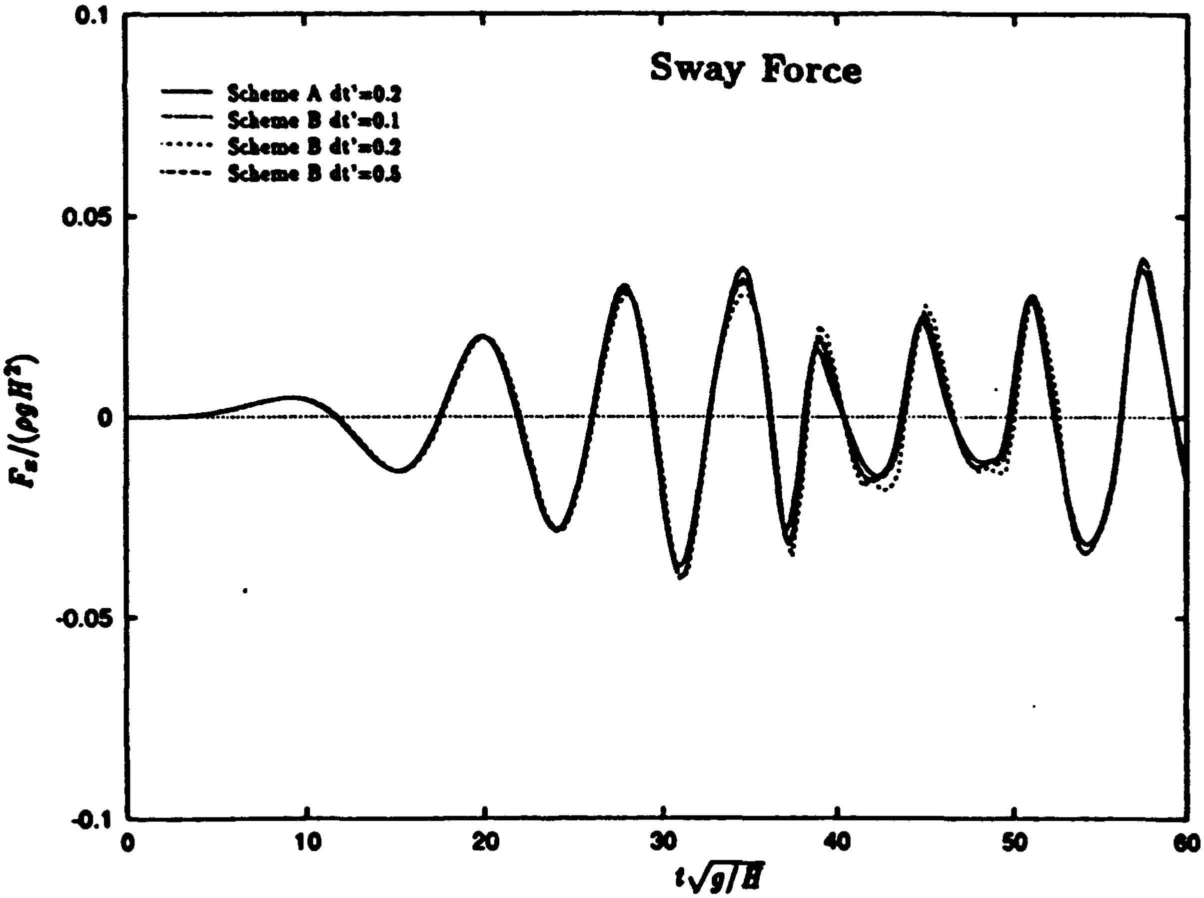


Effect of time step size on heave ($\varepsilon = 10^{-6}$)

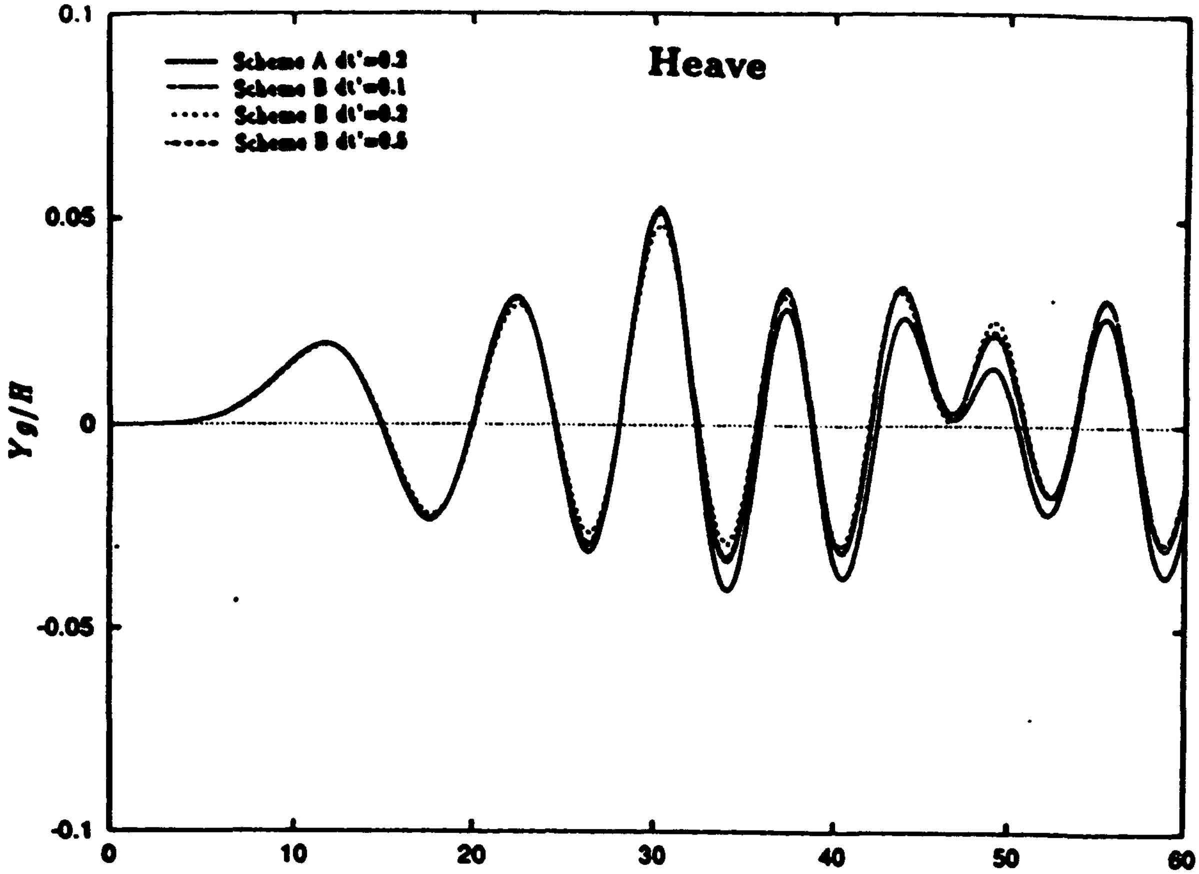


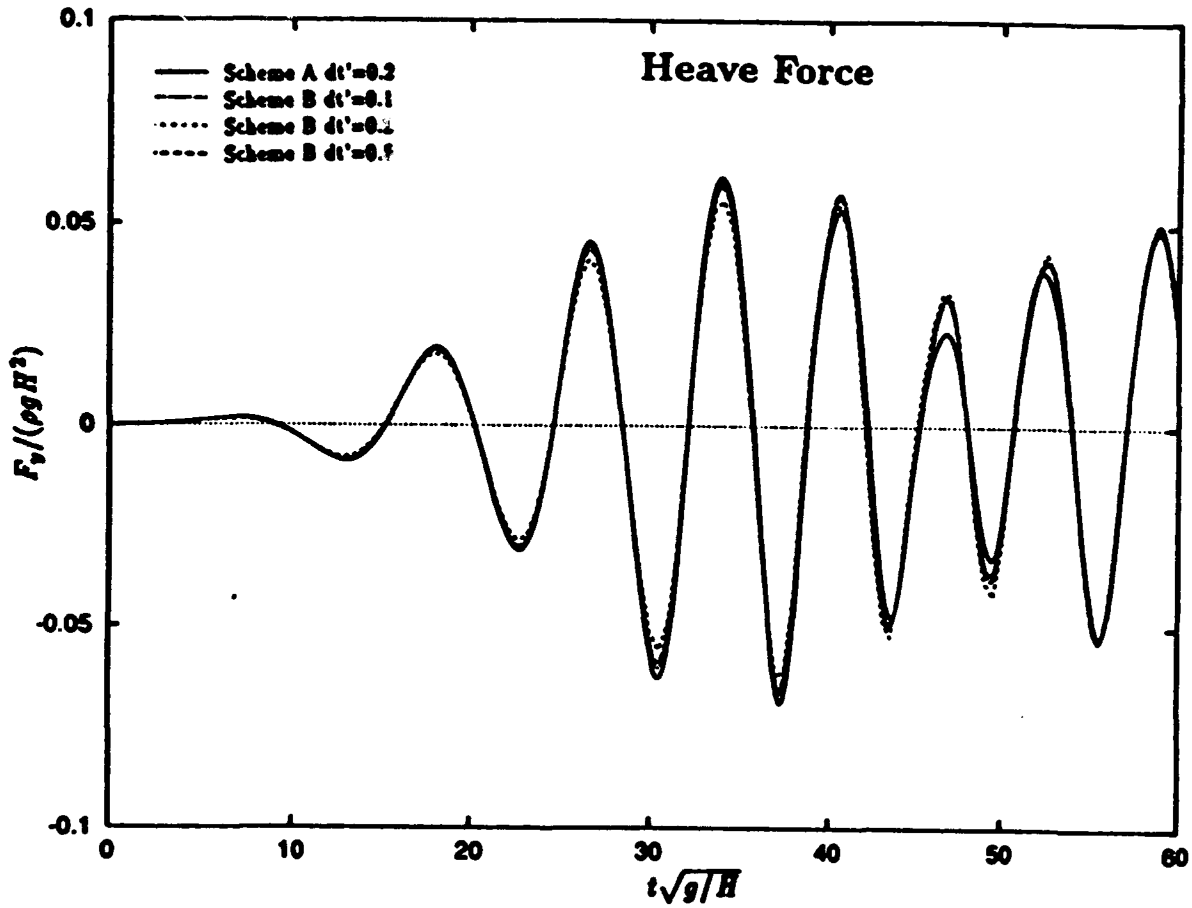
Effect of time step size on roll ($\varepsilon = 10^{-6}$)



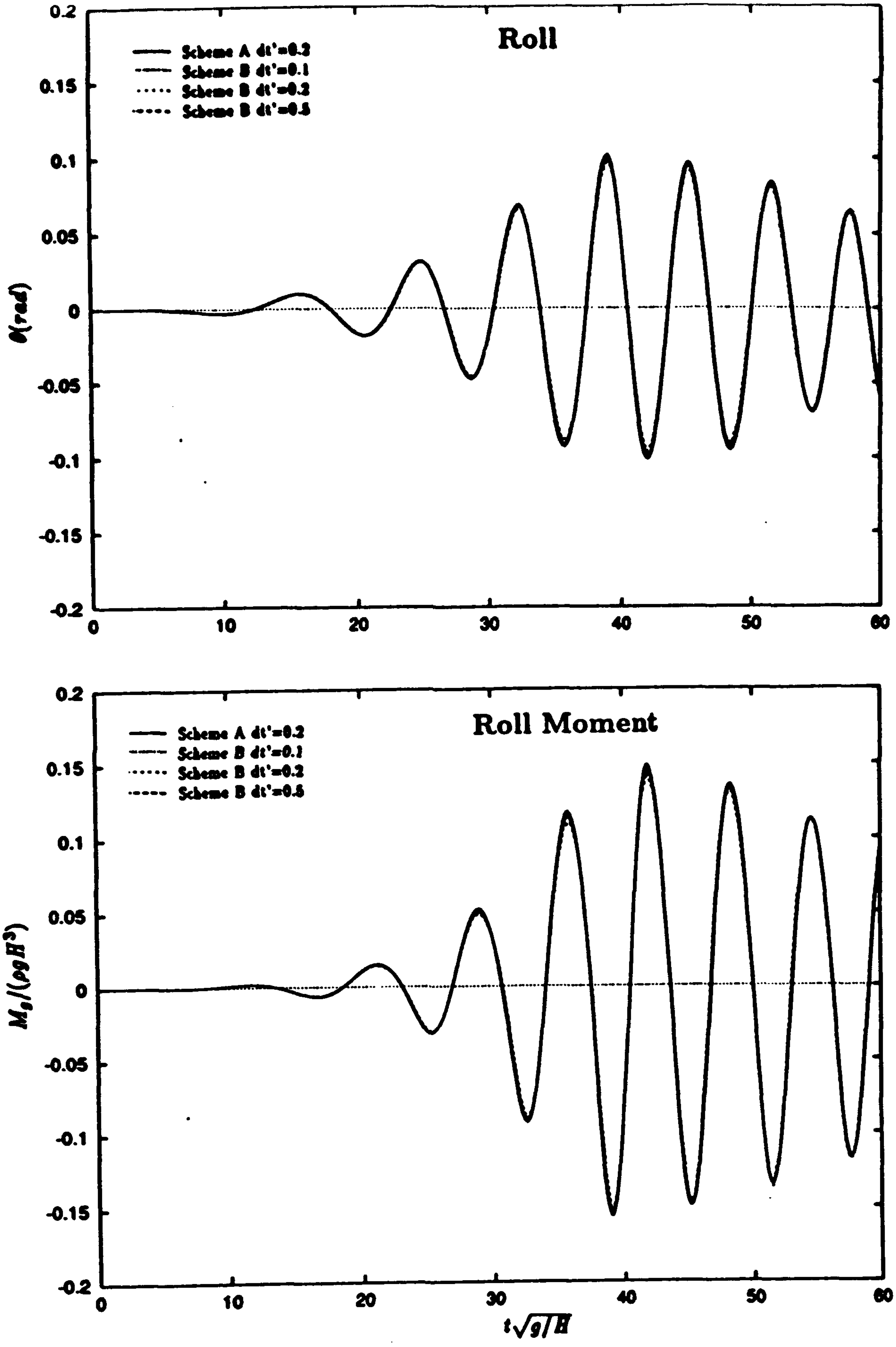


Comparisons of sway using the two $\frac{\delta\phi}{\delta t}$ schemes





Comparisons of heave using the two $\frac{\delta\phi}{\delta t}$ schemes



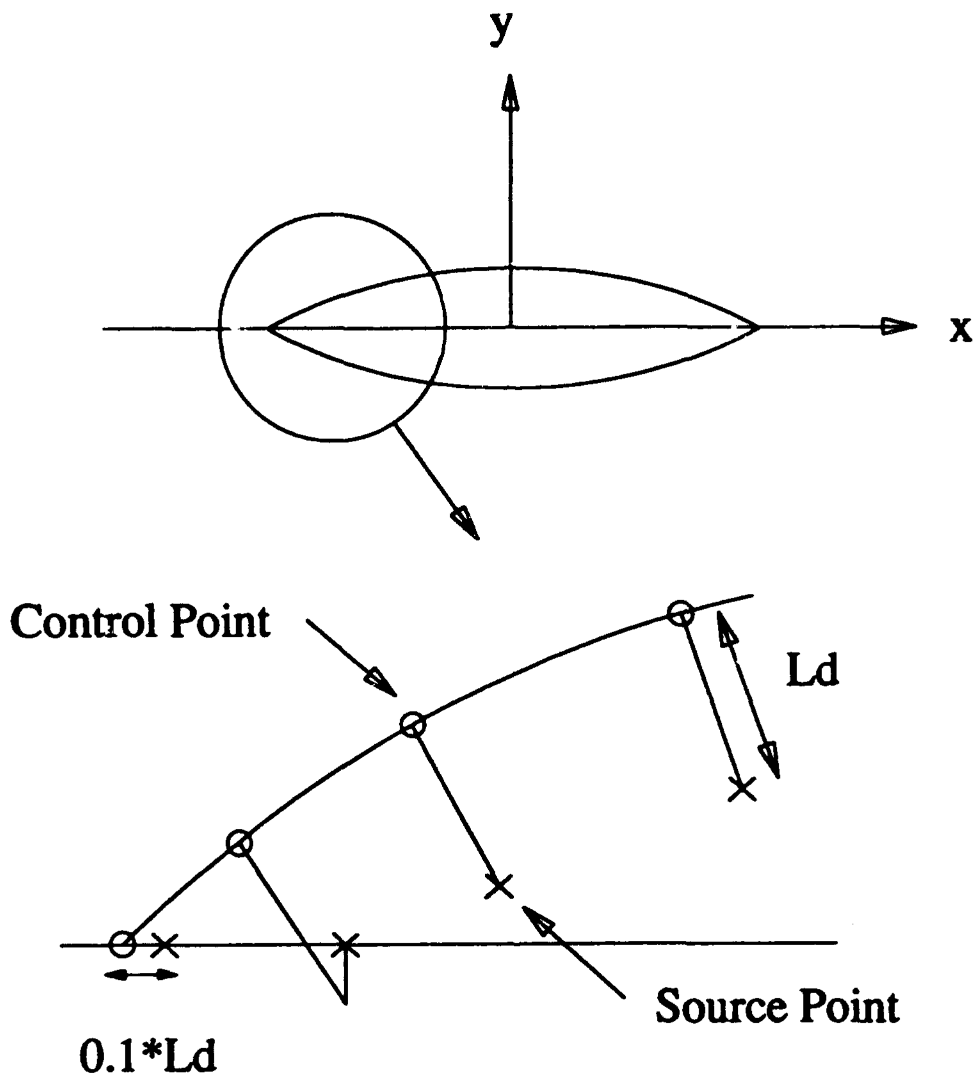
Comparisons of roll using the two $\frac{\delta\phi}{\delta t}$ schemes

Wigley Hull

$$y(x,z) = \frac{B}{2}\left(1-\left(\frac{2x}{L}\right)^2\right)\left(1-\left(\frac{z}{T}\right)^2\right)\left(1-a_2\left(\frac{2x}{L}\right)^2\right)$$

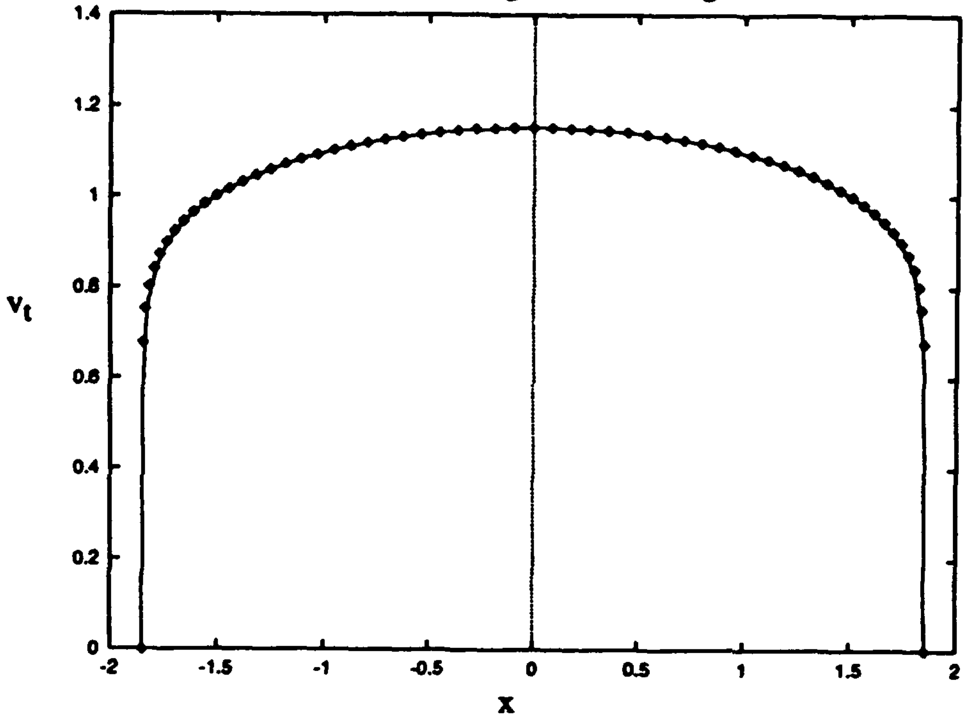
where
- L = model length
- B = model full beam
- T = model draft
- a_2 = coefficient for bow fullness
  - = 0.0, standard hull
  - = .2 for modified Wigley hull III

For both the standard hull and the modified hull III, L/B=10 and B/T=1.6.

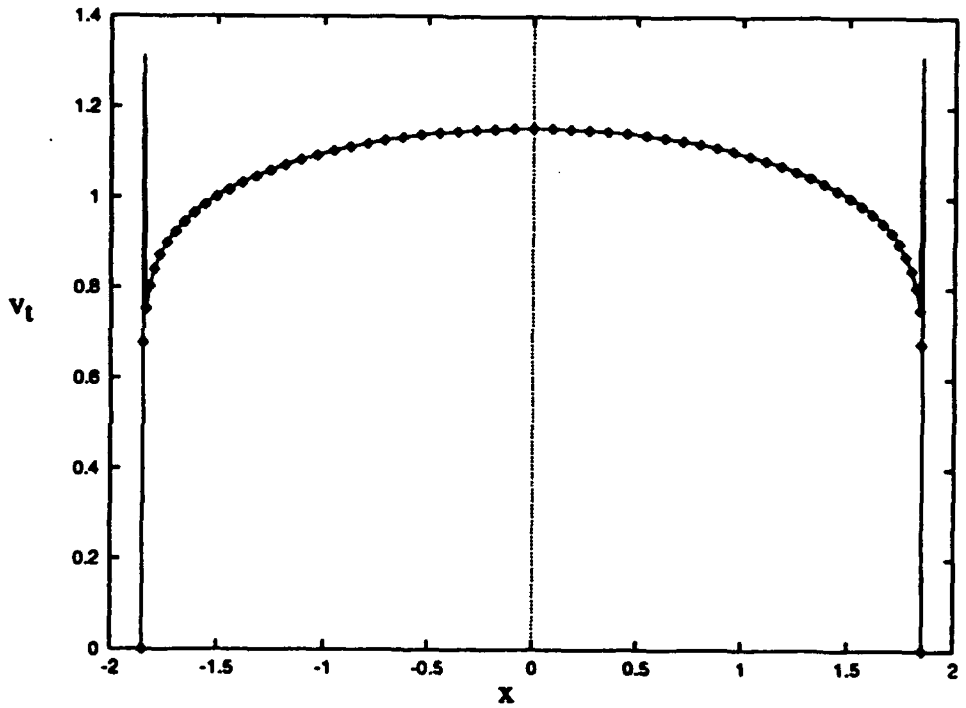


Desingularization near the leading edge of a Karman - Trefftz airfoil

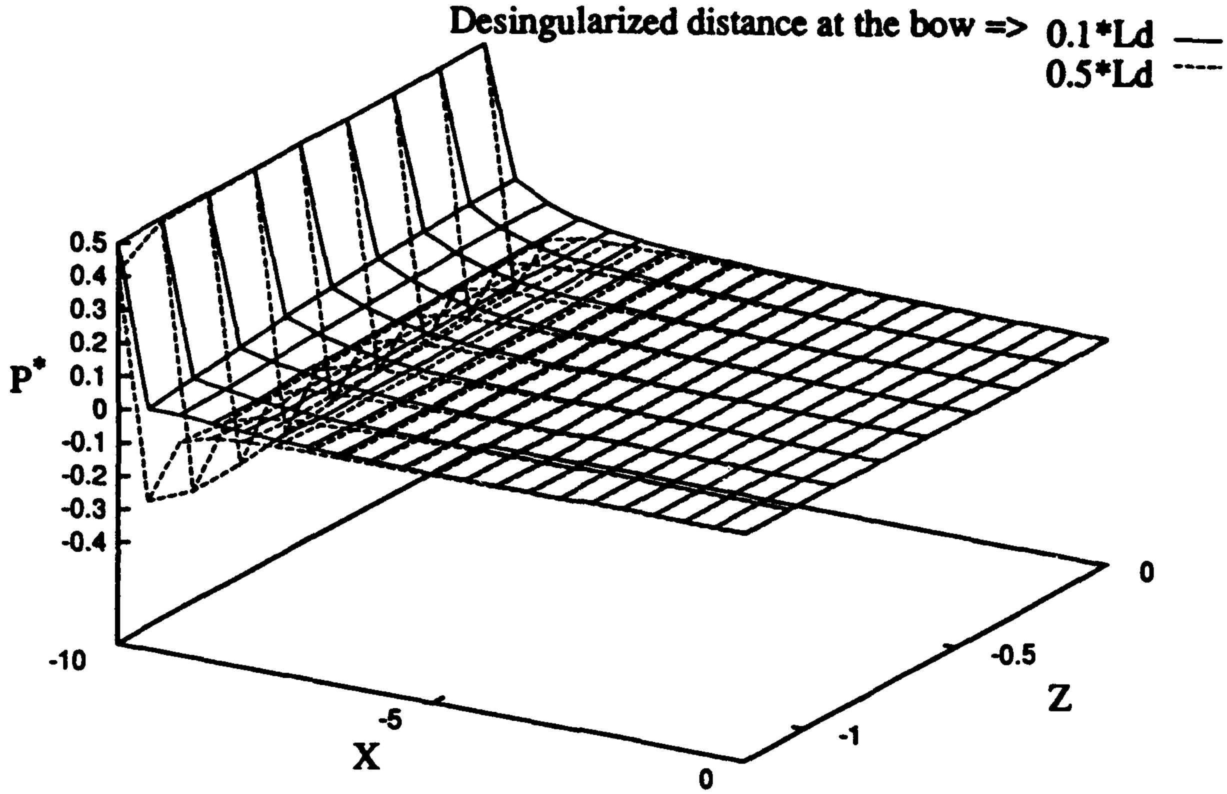
Desingularized distance at leading and trailing edge = 0.1*Ld
Entrance half angle = 13.5 degrees

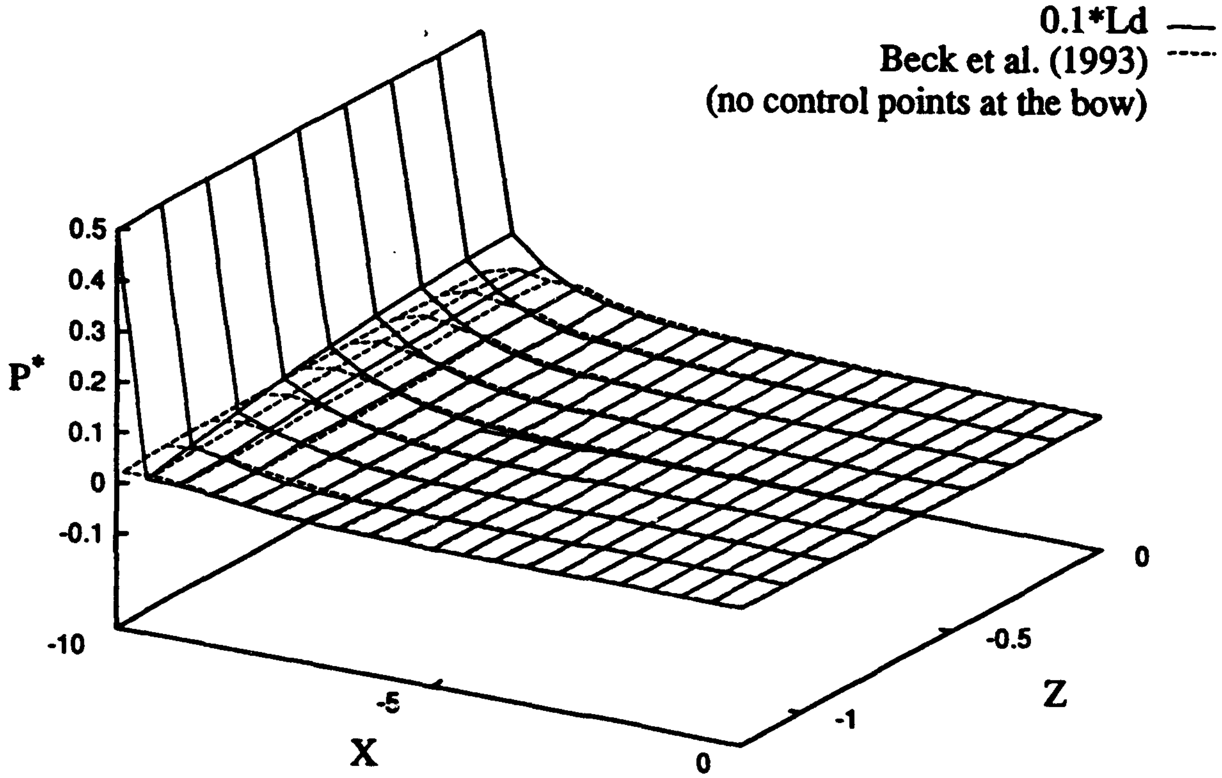


Desingularized distance at leading and trailing edge = 0.5*Ld
Entrance half angle = 13.5 degrees

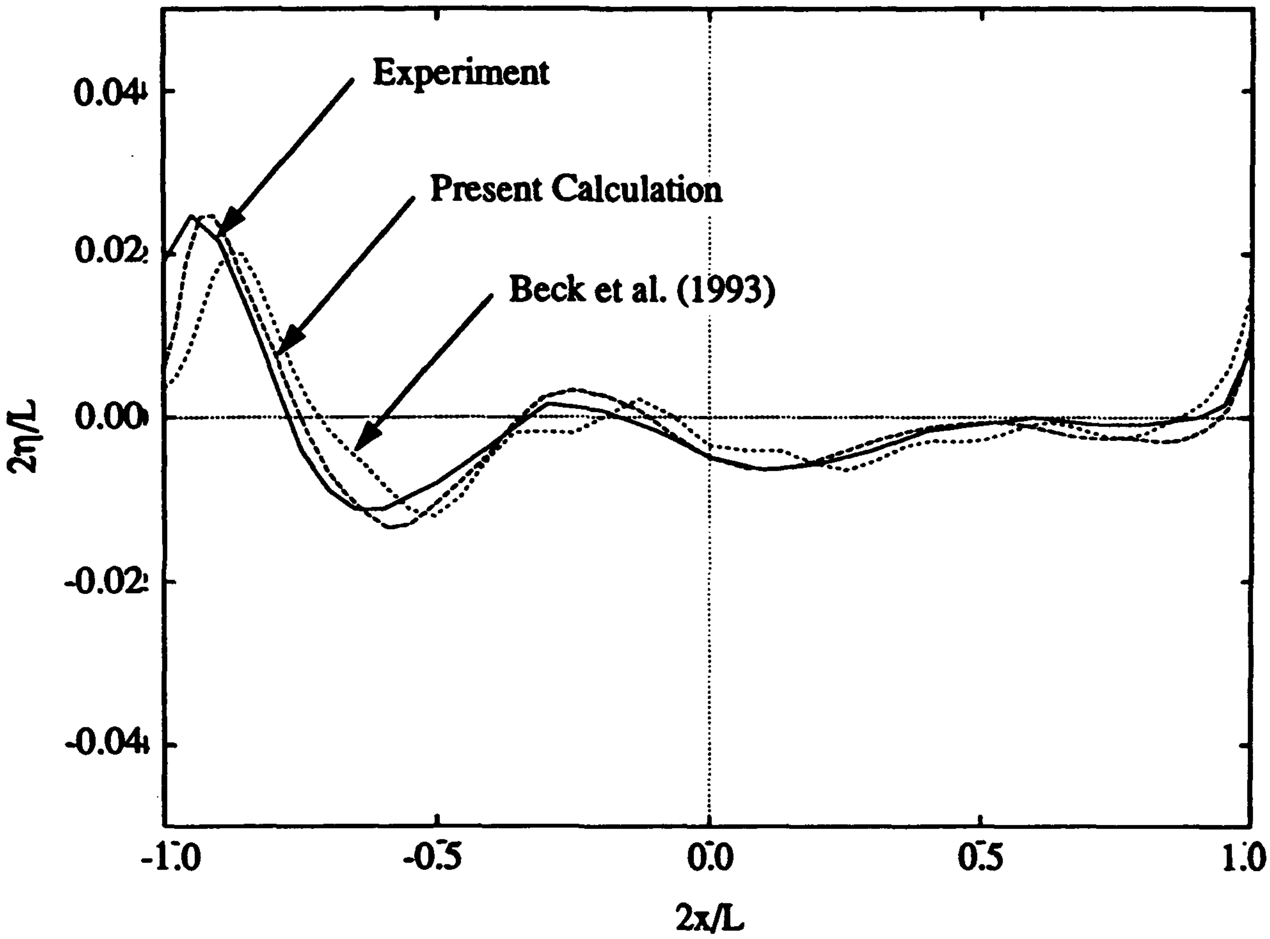


The effect of desingularized distance on surface tangential velocity (v_t) for a Karman - Trefftz airfoil

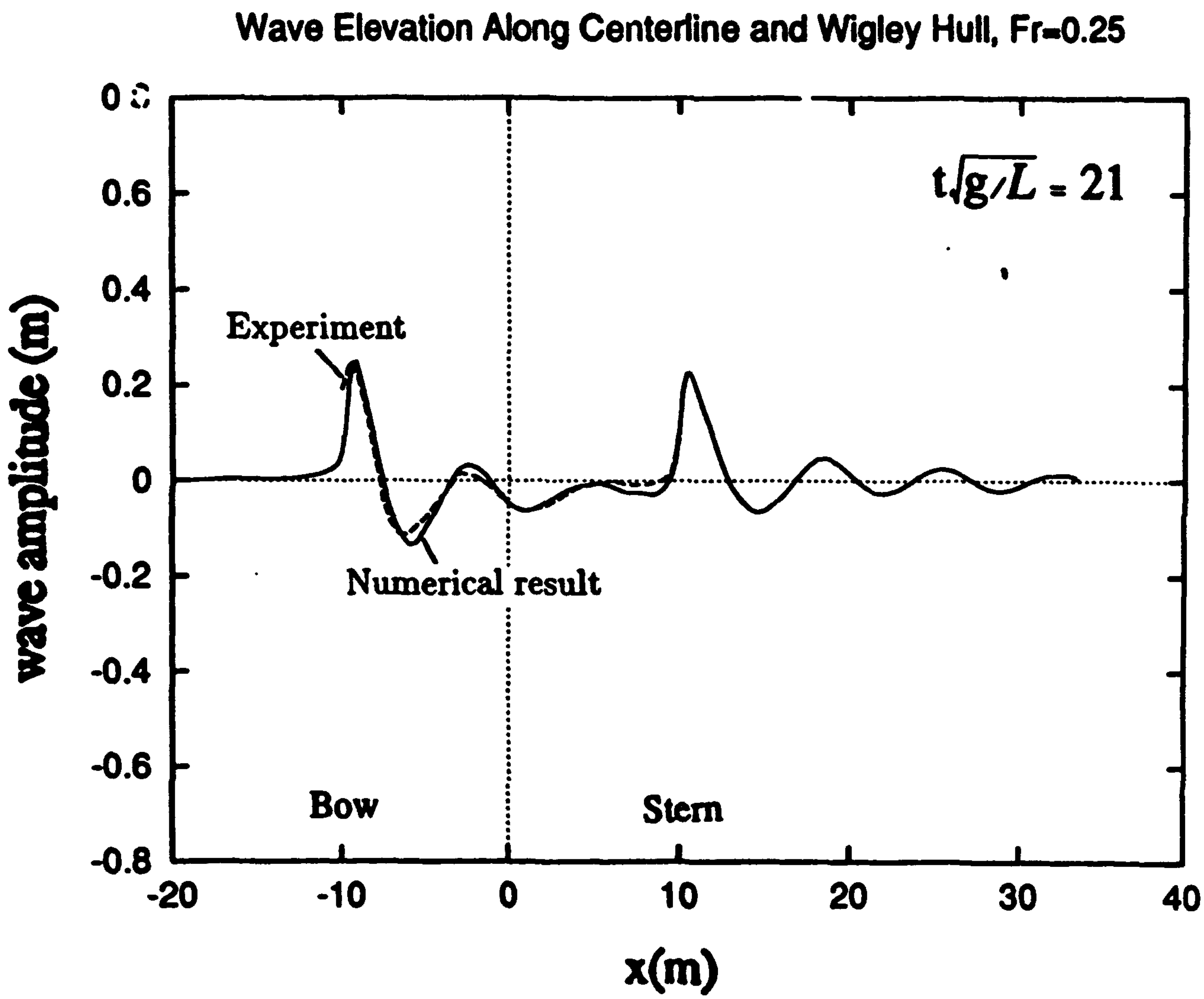




Wigley hull double body solution
pressure on the forward half of the body



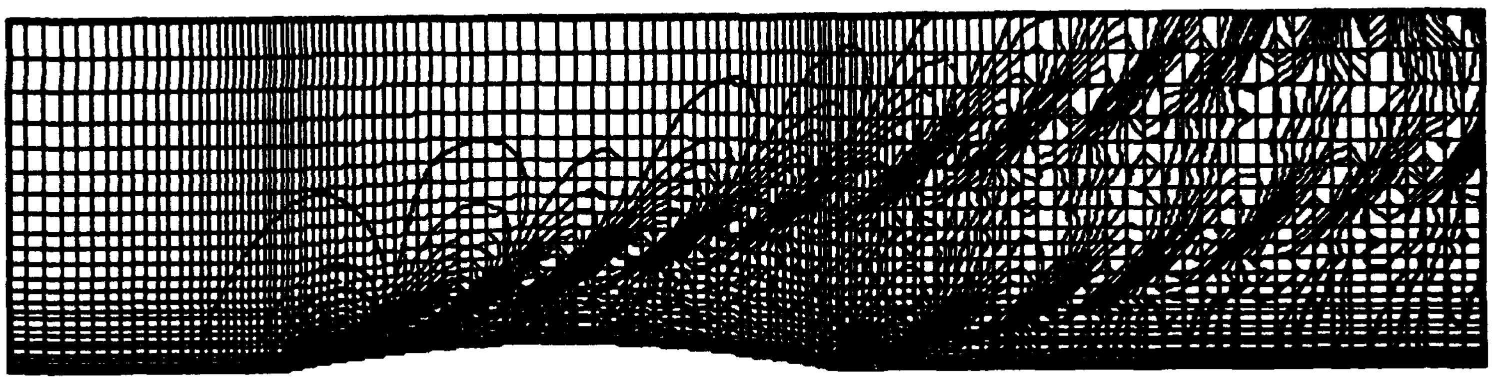
Wave profile along the standard Wigley hull
(Fr = 0.25, fixed sinkage and trim)



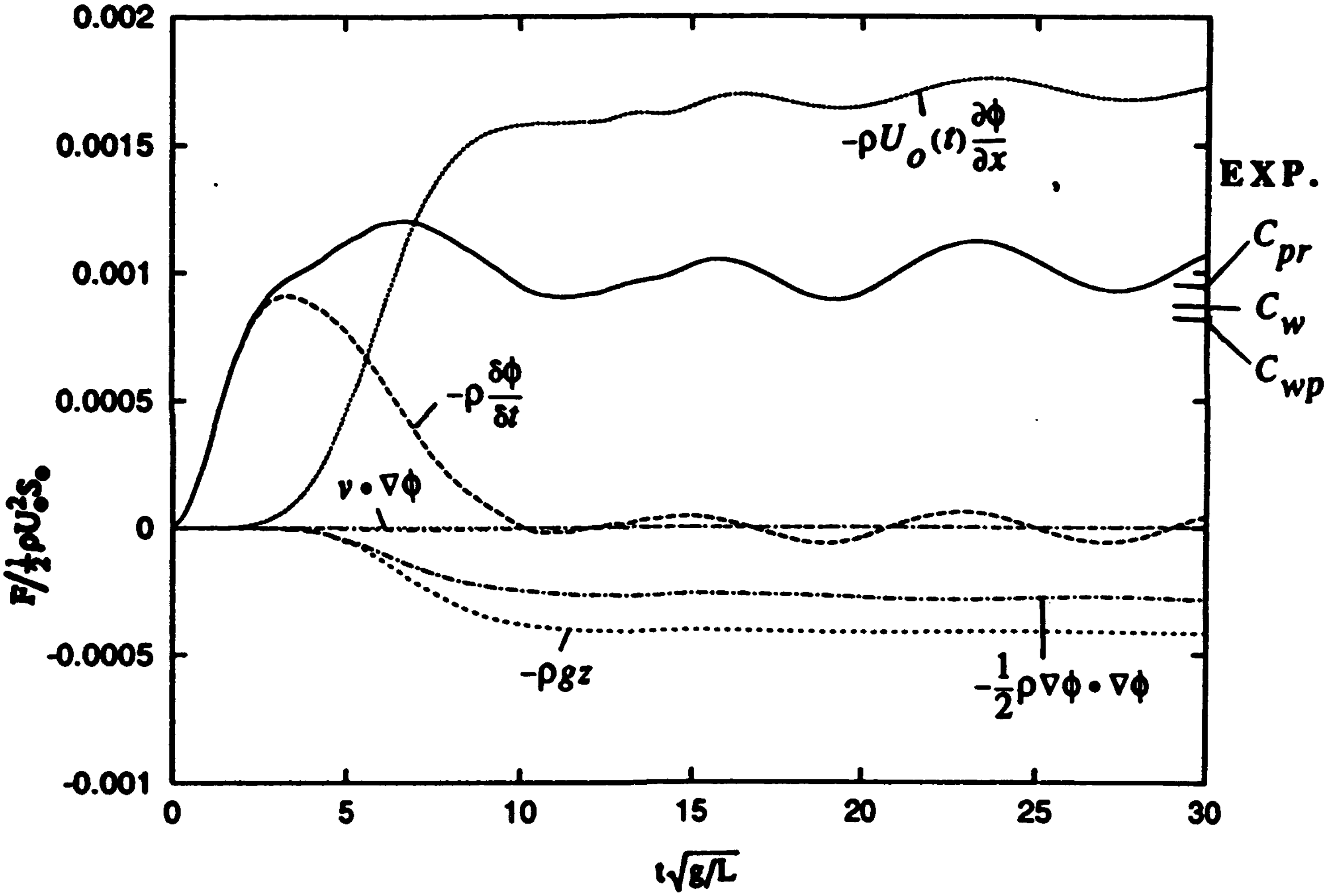
Wave Elevation Along Centerline and Wigley Hull, Fr=0.25
$t\sqrt{g/L} = 21$
Experiment
Numerical result
Bow
Stern
wave amplitude (m)
x(m)
0.6
0.4
0.2
0
-0.2
-0.4
-0.6
-0.8
-20
-10
0
10
20
30
40

Shaded rendering of the waves generated by Wigley Hull,
$Fr = 0.25$, $t\sqrt{g/L} = 21$, $N_{FS} = 2904, N_{body} = 612$

Contours of the waves generated by Wigley Hull,
$Fr = 0.25,\ t\sqrt{g/L} = 21,\ N_{FS} = 2904, N_{body} = 612$



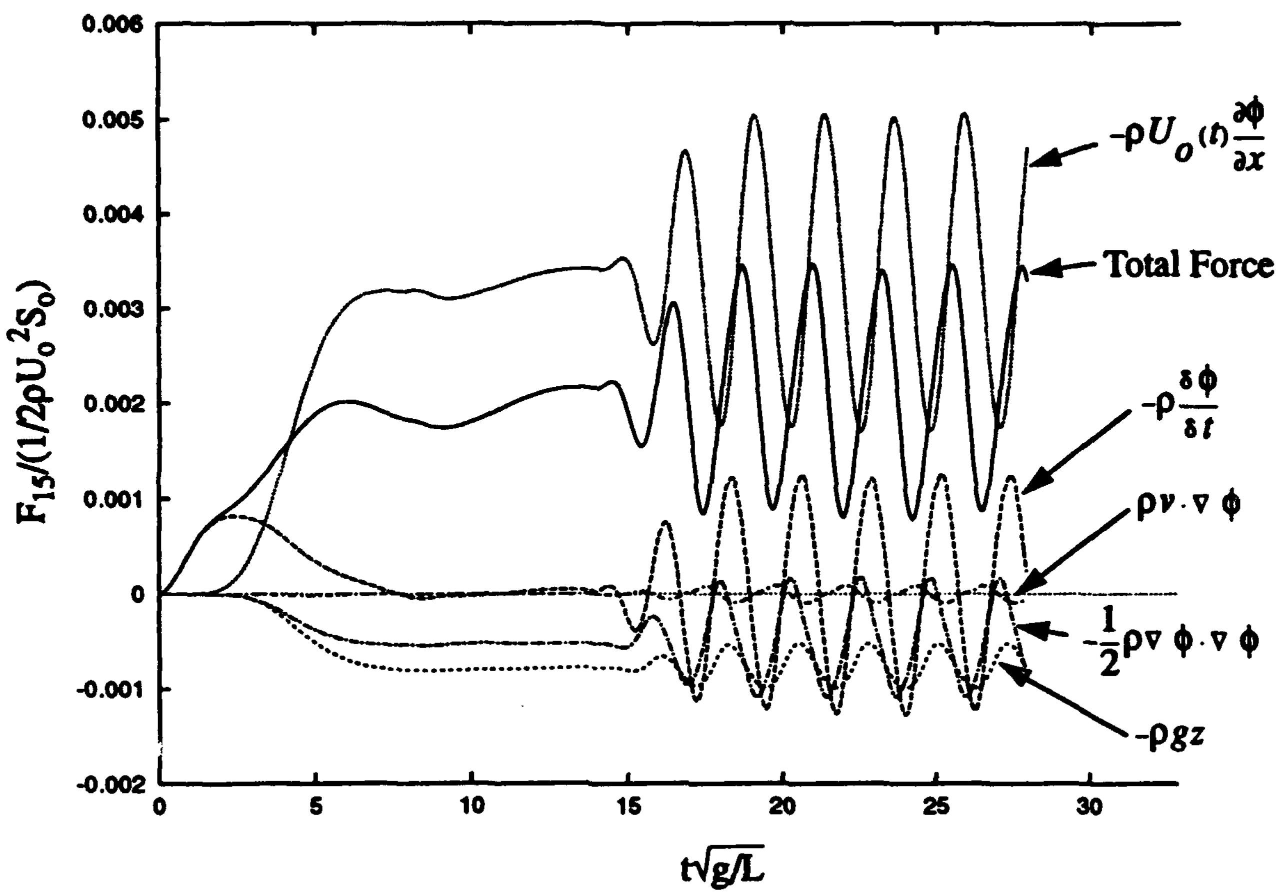
Wave Resistance Components for Wigley Hull, Fr=0.25



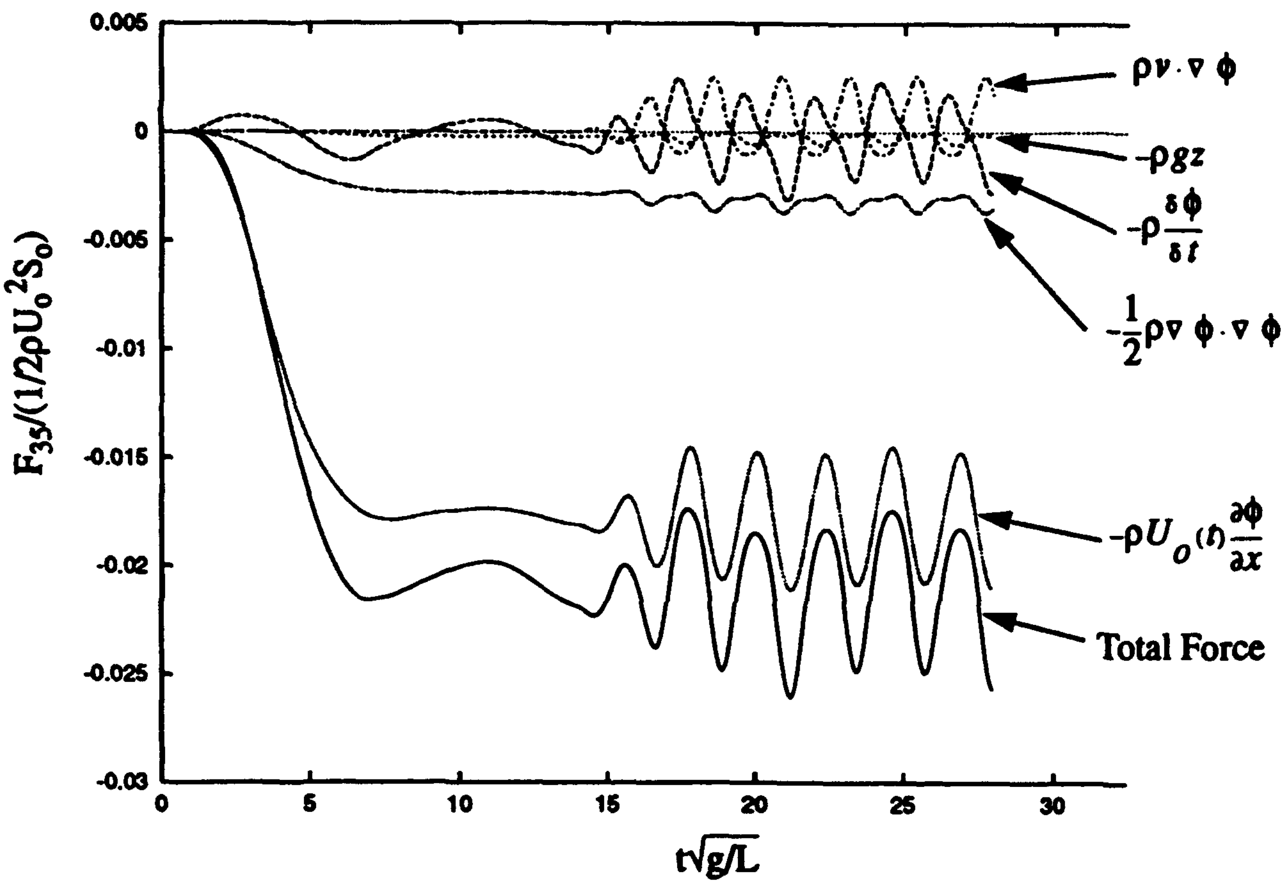
Added Mass and Damping in Heave and Pitch

for

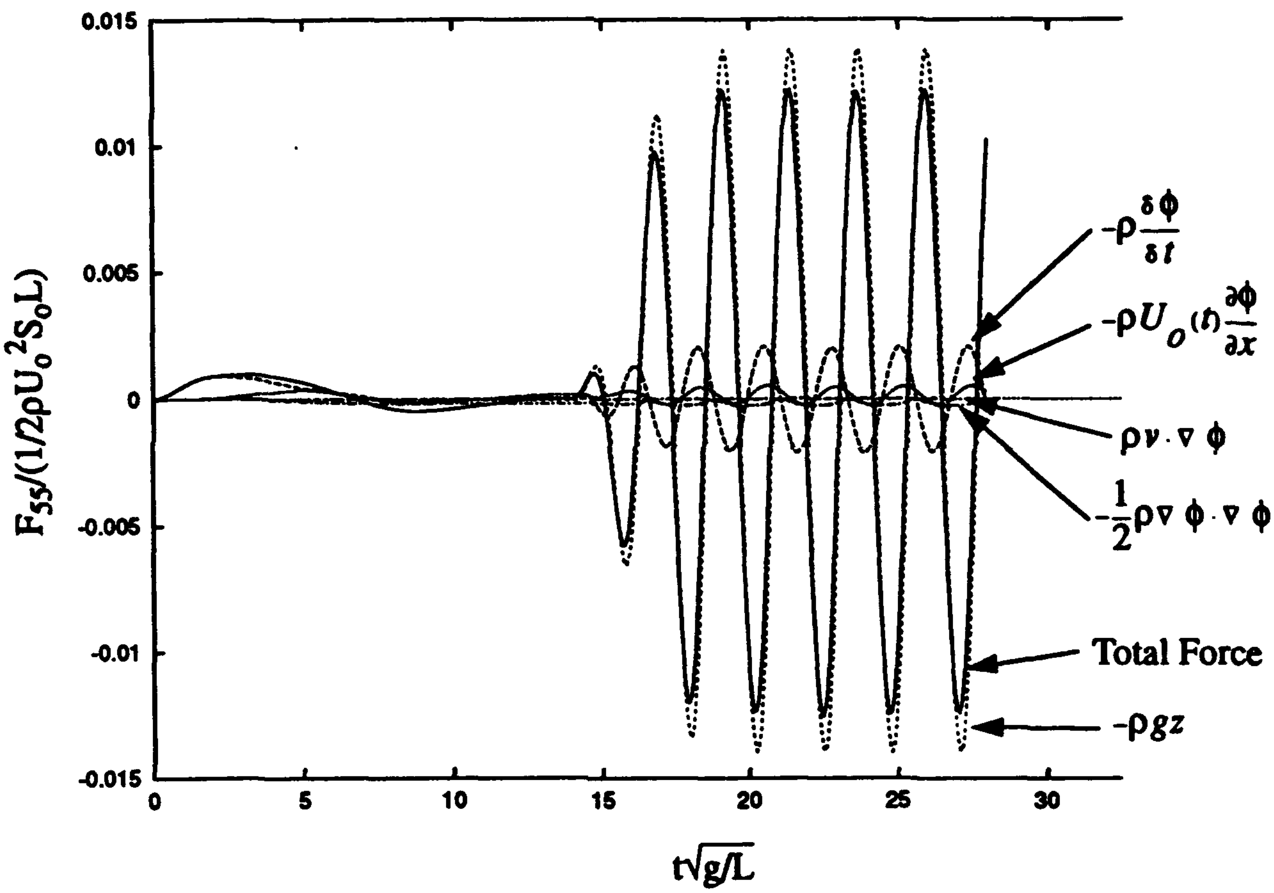
Modified Wigley Hull III



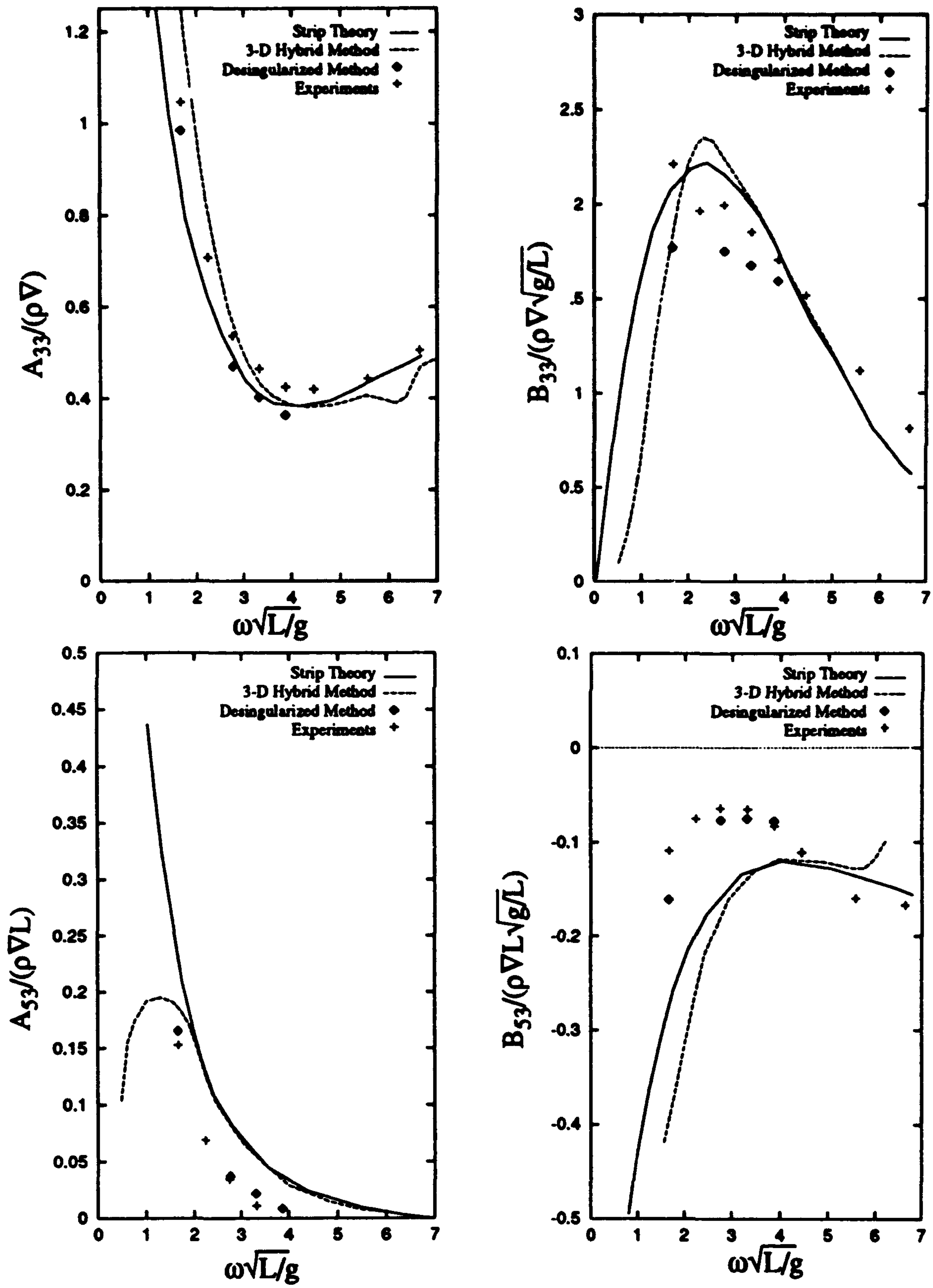
Modified Wigley hull III - surge force due to pitch excitation
Fr = 0.3, pitch amplitude = 1.5°, $\omega/\sqrt{g/L}$ = 2.76642



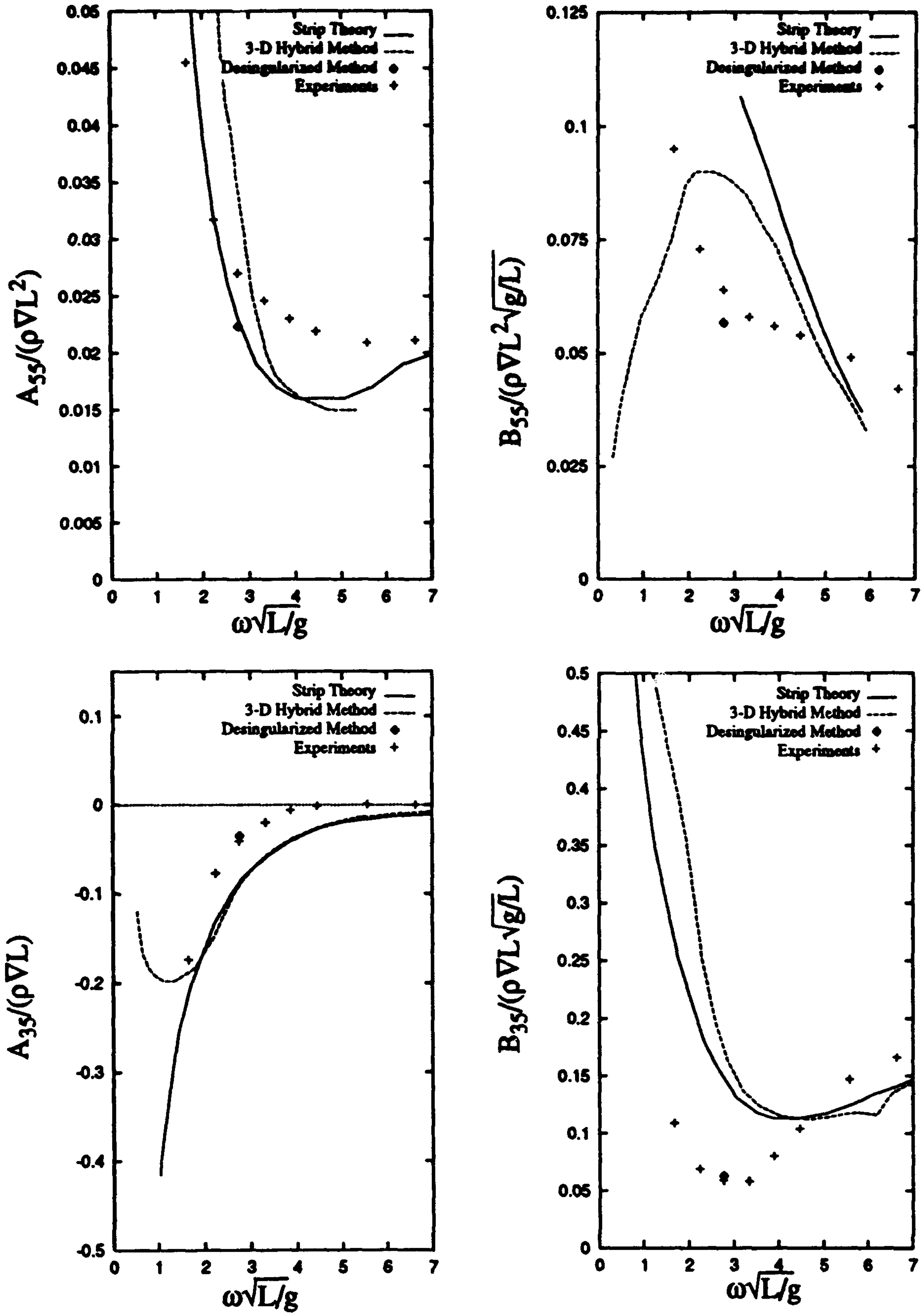
Modified Wigley hull III - heave force due to pitch excitation
Fr = 0.3, pitch amplitude = 1.5°, $\omega/\sqrt{g/L}$ = 2.76642



Modified Wigley hull III - pitch force due to pitch excitation
Fr = 0.3, pitch amplitude = 1.5°, $\omega/\sqrt{g/L}$ = 2.76642



Comparisons of experimental and theoretical added mass and damping coefficients for forced heave
(Wigley model III, Fr = 0.3, z_a/L= 0.00833)

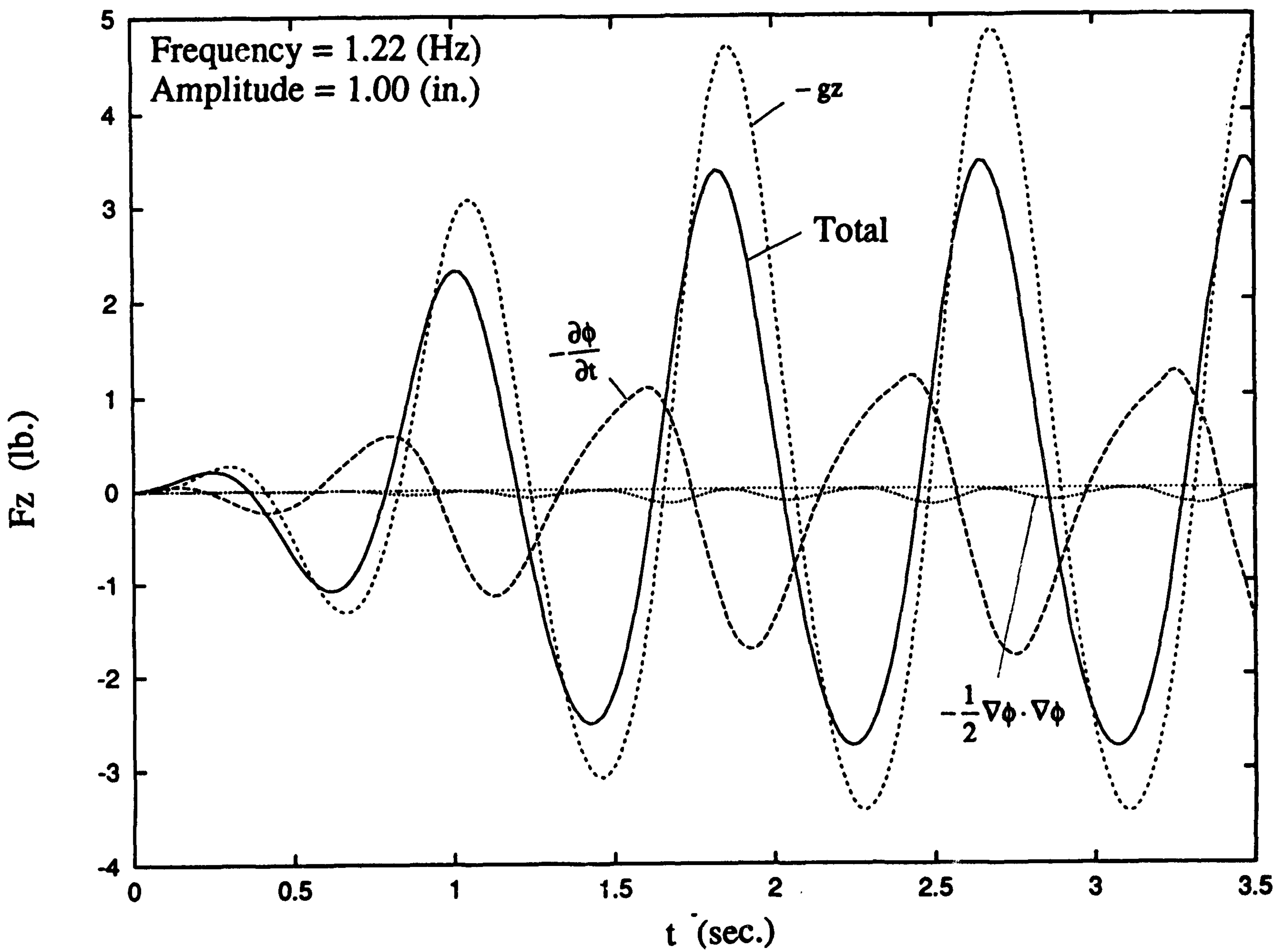


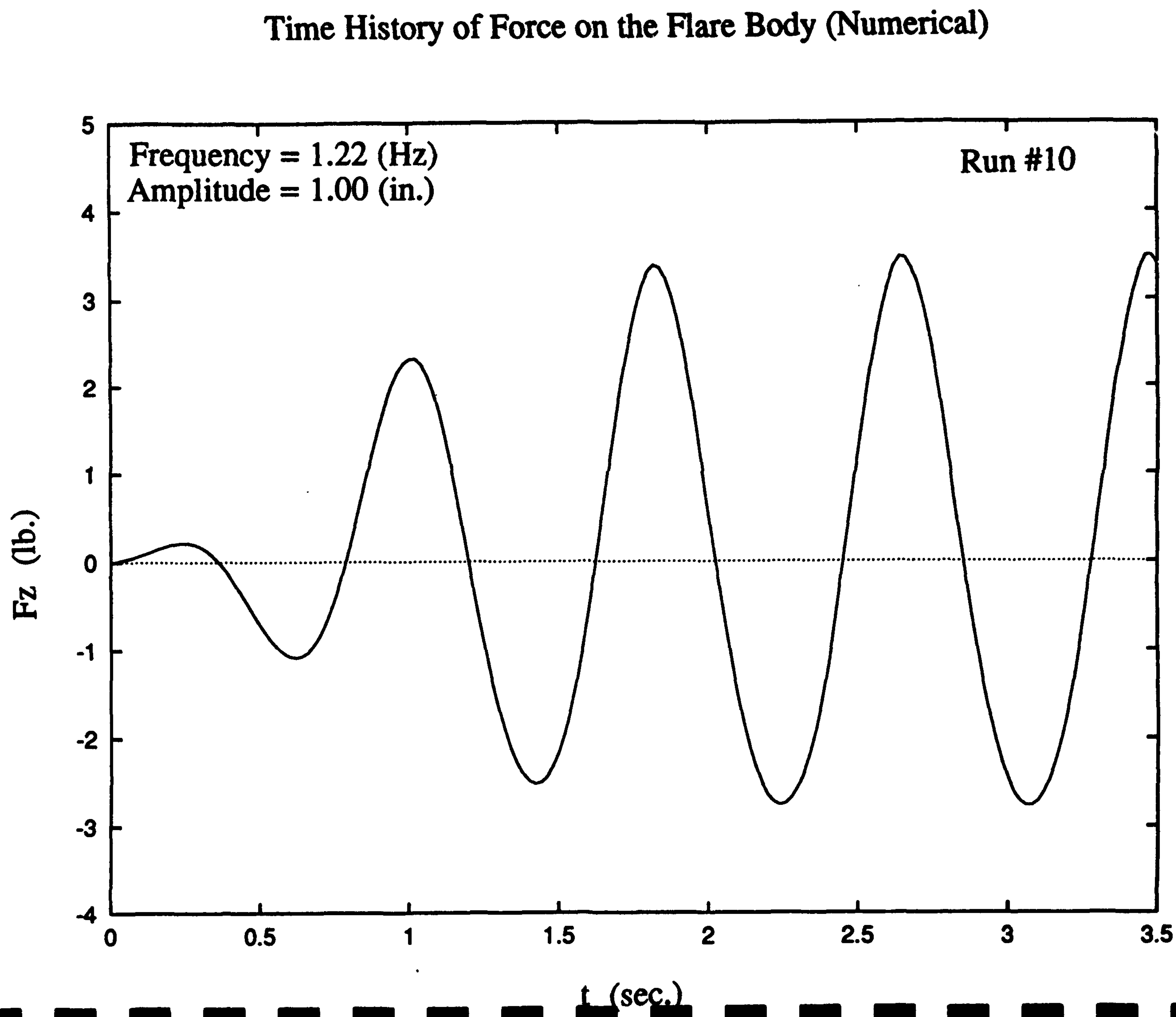
Comparisons of experimental and theoretical added mass and damping coefficients for forced pitch
(Wigley model III, Fr = 0.3, pitch amplitude = 1.5°)

Forced Oscillation
of
Large-Flare Body
$U_o = 0$

Time History of Force on the Flare Body (Numerical)

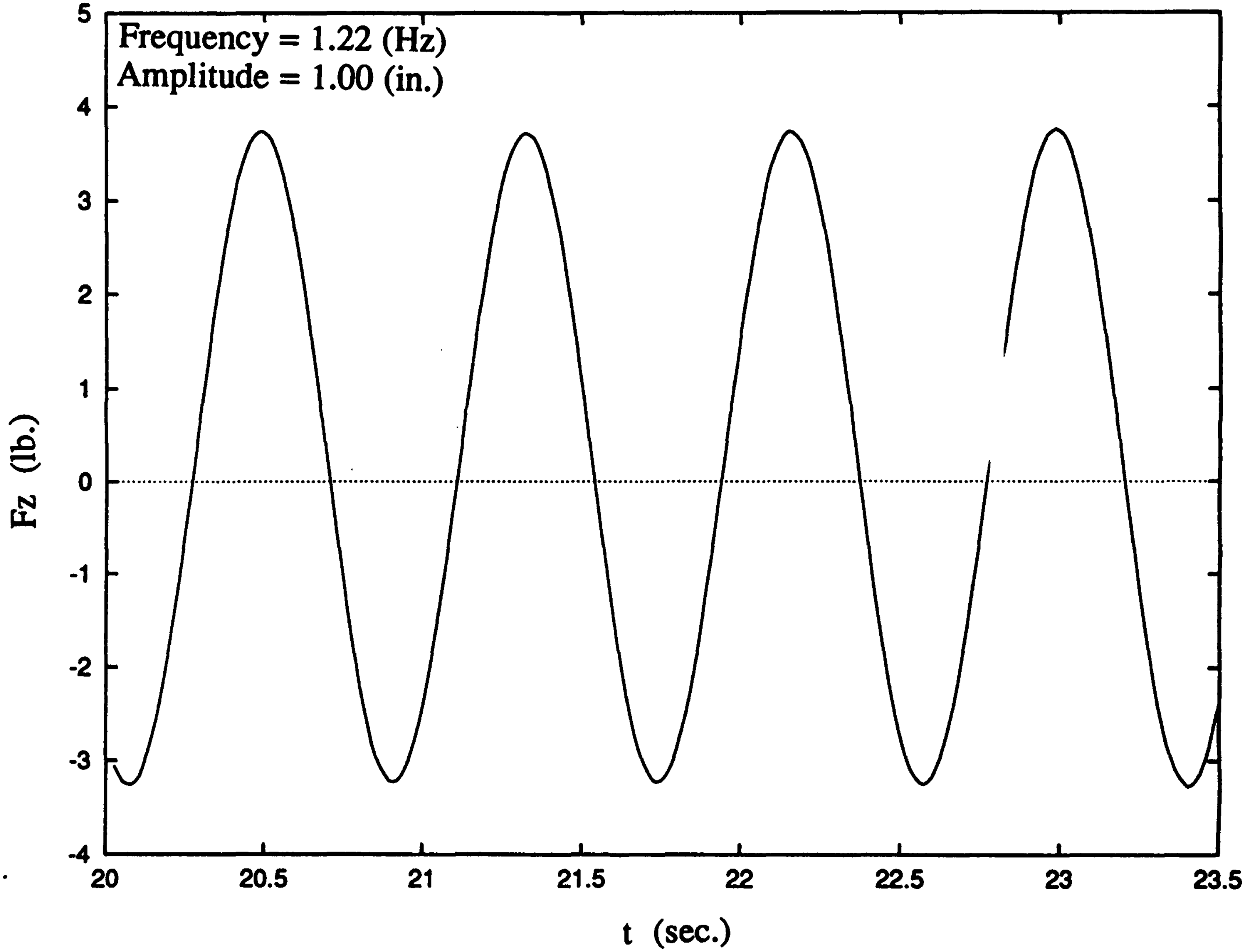
Run #10




Time History of Force on the Flare Body (Numerical)
Frequency = 1.22 (Hz)
Amplitude = 1.00 (in.)
Run #10
Fz (lb.)
t (sec.)
5
4
3
2
1
0
-1
-2
-3
-4
0
0.5
1
1.5
2
2.5
3
3.5

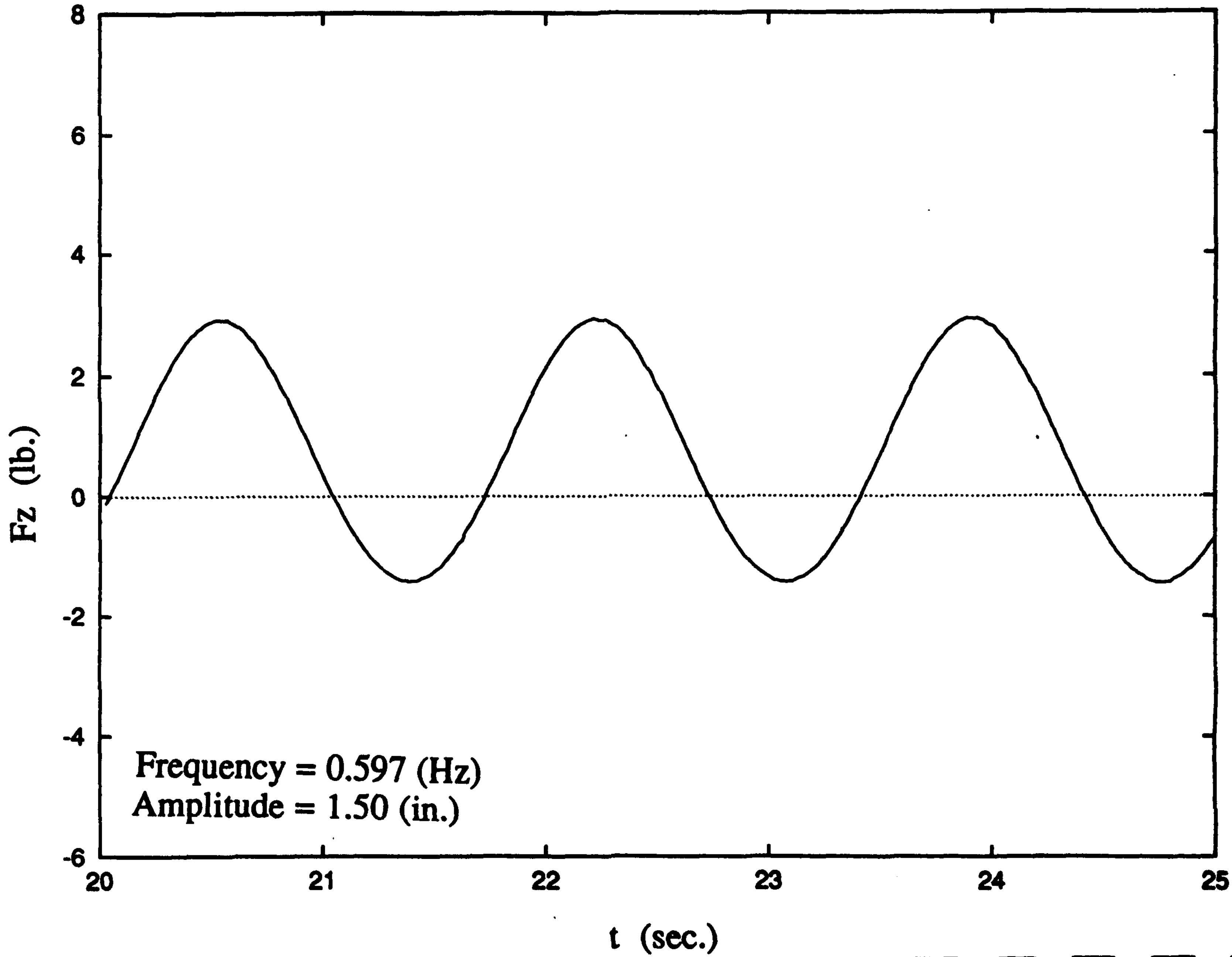
Time History of Force on the Flare Body (Experiment)

Run #10



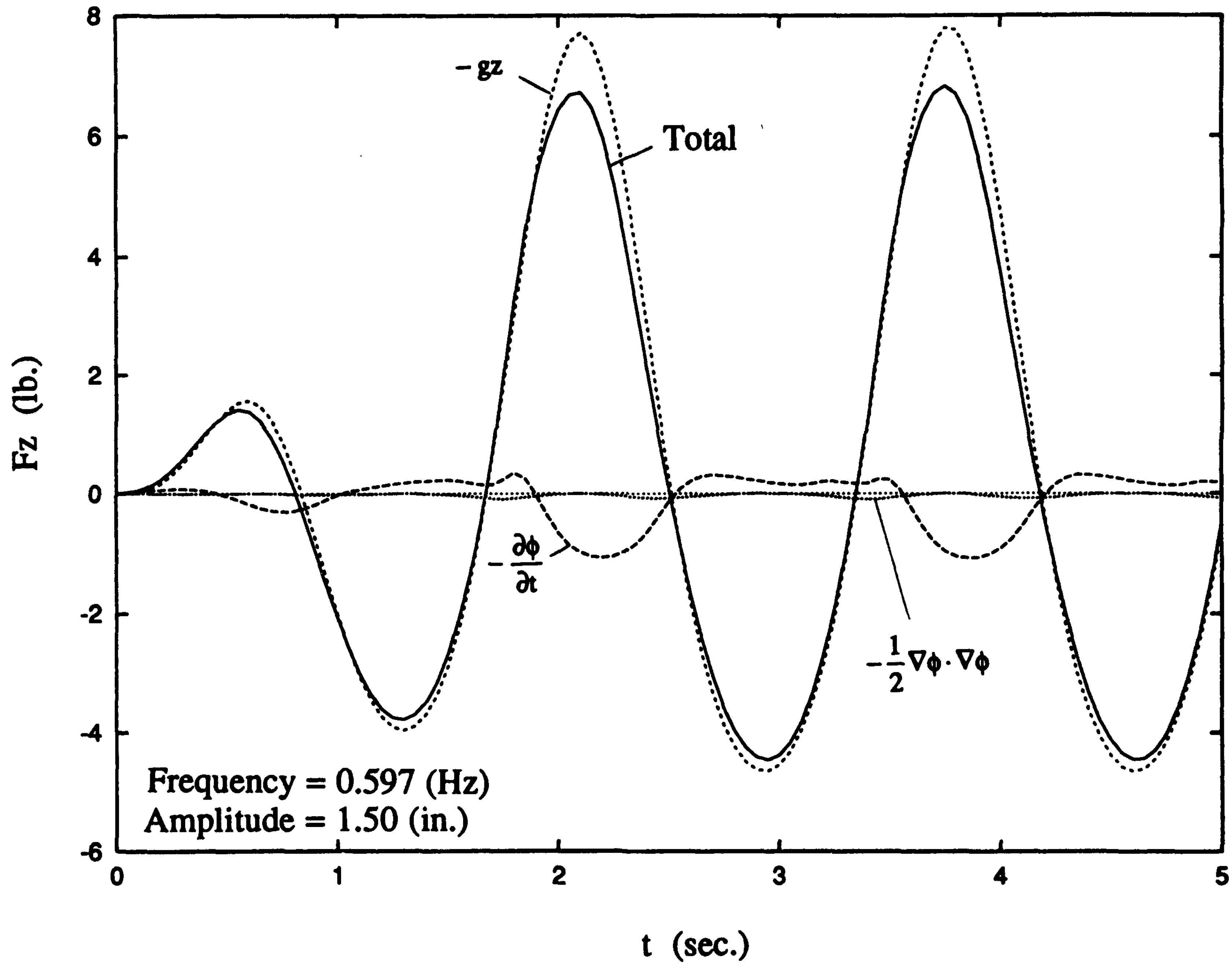
Time History of Force on the Flare Body (Experiment)

Run #16



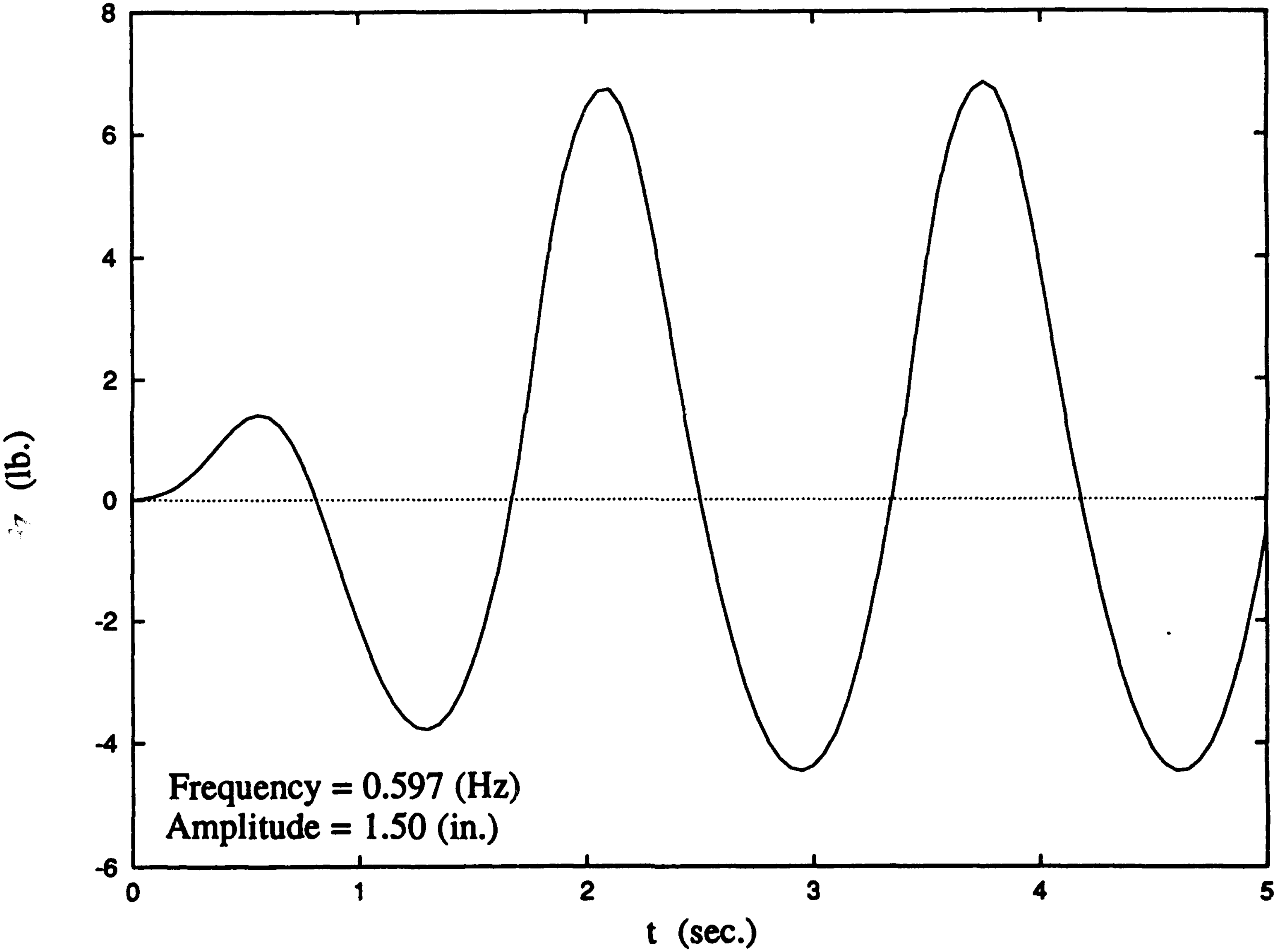
Time History of Force on the Flare Body (Numerical)

Run #16



Time History of Force on the Flare Body (Numerical)

Run #16



LOADS ASSOCIATED WITH THE HYDRODYNAMIC IMPACT OF FLAT WEDGES (flat cylinders)

William S. Vorus

The University of Michigan

a) Zero Viscosity

b) Zero Compressibility

c) Zero Gravity

Hydrodynamic Impact of:

1) Relatively Flat Cylinders (2D)

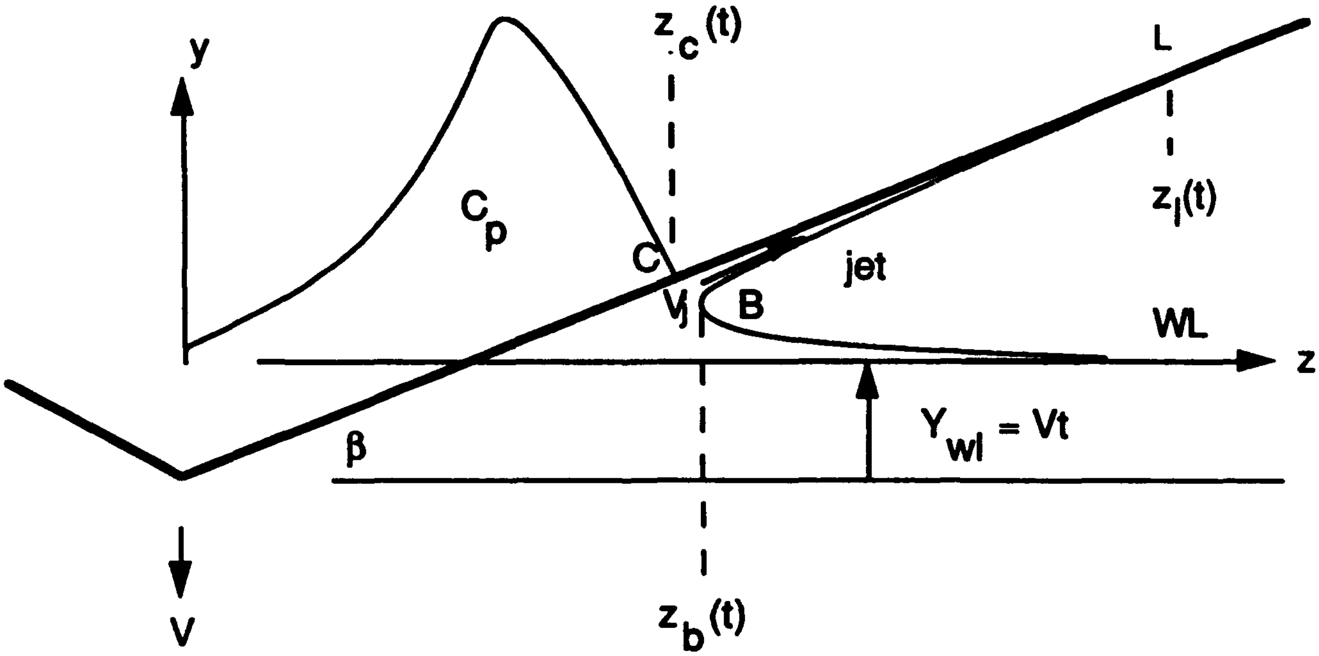
2) of Otherwise Arbitrary (smooth) Geometry

3) Including Variable Impact Velocity
(with oscillatory perturbation)

4) Time Varying Contour Shape

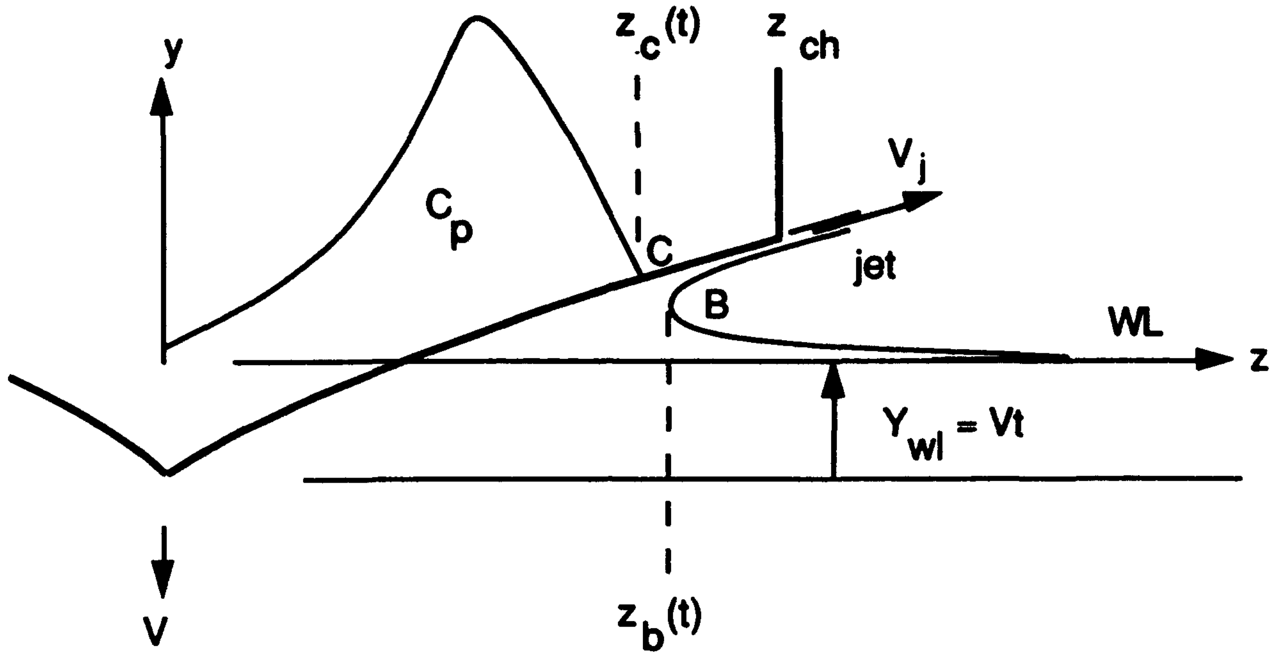
5) Multiple Contours

DEMONSTRATION IN TERMS OF SELF-SIMILAR SEMI-INFINITE WEDGE IMPACT



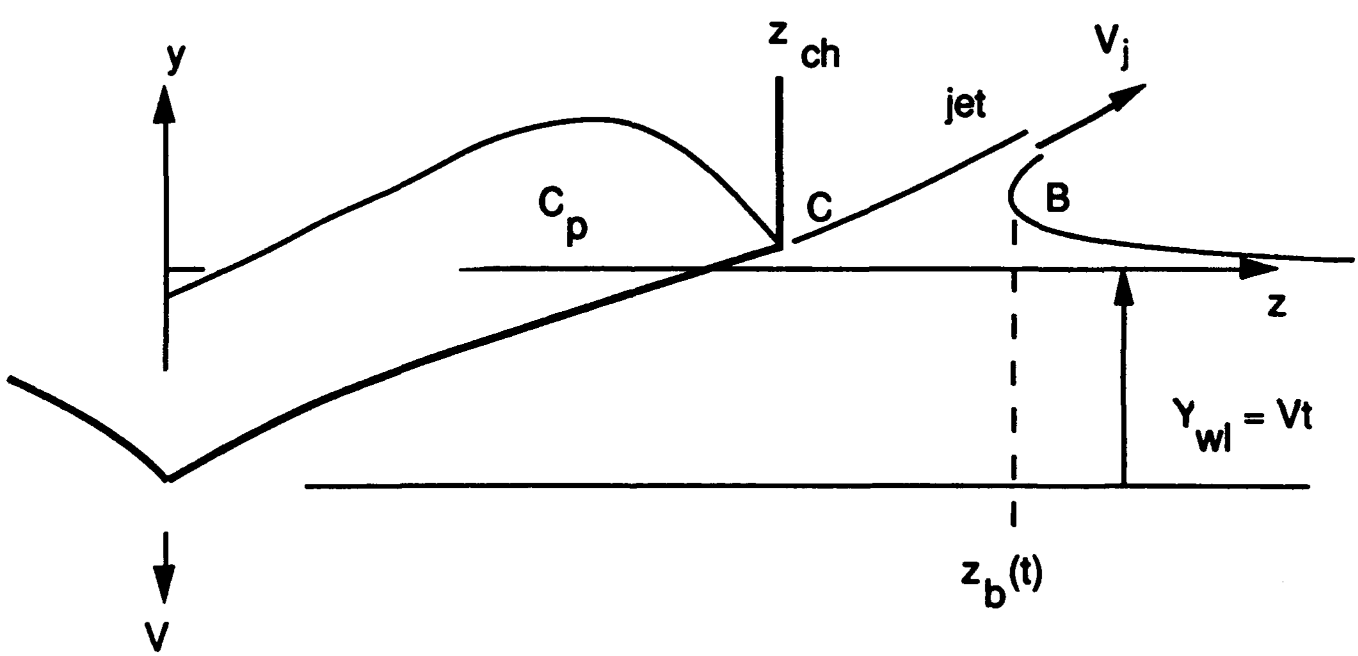
WEDGE IMPACT FLOW

FIGURE 1



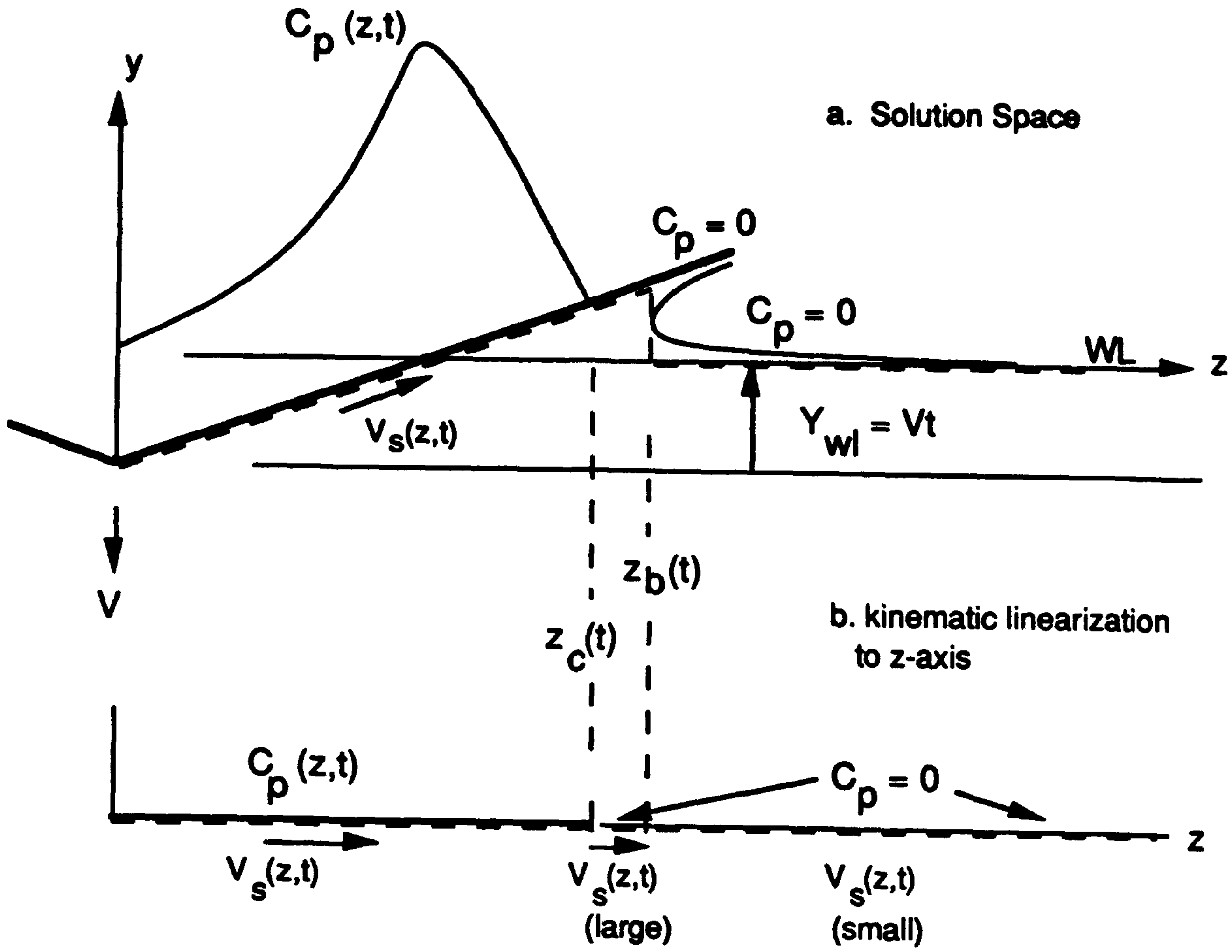
CYLINDER IMPACT (CUW)

FIGURE 1a



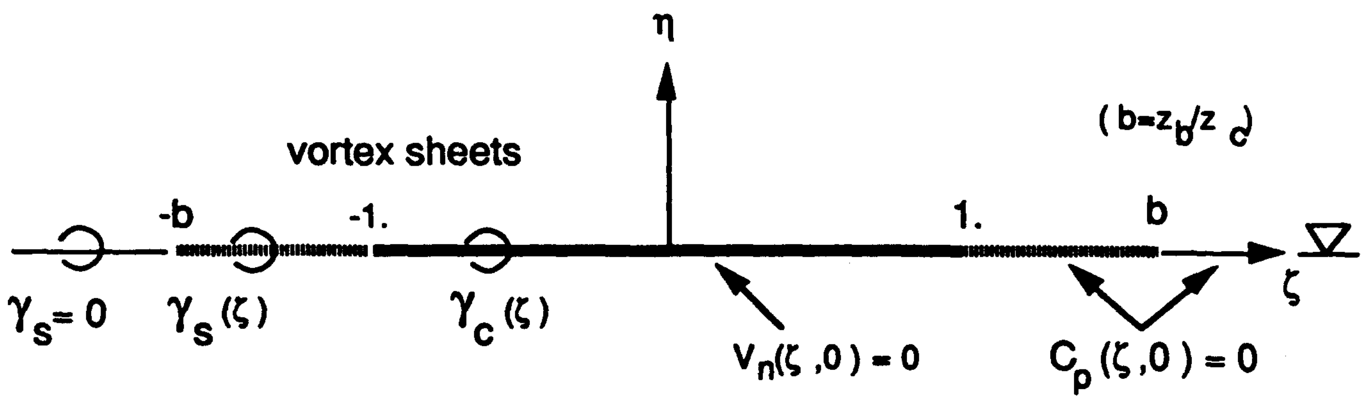
CYLINDER PENETRATION (CW)

FIGURE 1b



PHYSICAL APPROXIMATION

FIGURE 2



MATHEMATICAL MODEL

FIGURE 4

$$C_p(\zeta) = 0 \qquad \text{on} \qquad \zeta \geq 1$$

$$C_p(\zeta) = 1 - V_n^2(\zeta) - V_s^2(\zeta) + 2z_{ct}\left[\int_{\zeta_0=\zeta}^{b} V_s(\zeta_0)d\zeta_0 + \zeta V_s(\zeta)\right]$$

$$0 = 1 - V_s^2(\zeta) + 2z_{ct}\left[\int_{\zeta_0=\zeta}^{b} V_s(\zeta_0)d\zeta_0 + \zeta V_s(\zeta)\right] \qquad 1 \leq \zeta \leq b$$

This condition is clearly satisfied for $V_s(\zeta) = V_j$, a constant in $1 \leq \zeta \leq b$, giving:

$$z_{bt} = \frac{V_j^2 - 1}{2V_j}$$

with $z_{bt} \equiv z_{ct}b$,

and $V_j = -2\gamma_s$

WEDGE CONTOUR KINEMATIC BOUNDARY CONDITION

$$\frac{1}{2}\gamma_c(\zeta)\sin\beta + \frac{1}{2\pi}\int_{\zeta_0=-1}^{1}\frac{\gamma_c(\zeta_0)}{\zeta_0-\zeta}d\zeta_0 = -1 - \frac{\gamma_s}{\pi}\ln\sqrt{\frac{b^2-\zeta^2}{1-\zeta^2}} \qquad 0 \le \zeta \le 1$$

Solution:

$$\gamma_c(\zeta) = -\frac{2\zeta\cos\tilde{\beta}}{\sqrt{1-\zeta^2}}\left(\frac{1-\zeta^2}{\zeta^2}\right)^{\frac{\tilde{\beta}}{\pi}} \cdot$$

$$\cdot\left\{1 + \frac{\gamma_s}{\pi}\frac{(b^2-1)^\lambda}{2\lambda}\left[F\left(\lambda,\lambda,\lambda+1;1-b^2\right) - \left(\frac{1-\zeta^2}{b^2-\zeta^2}\right)^\lambda F\left(\lambda,\lambda,\lambda+1;\frac{\zeta^2(b^2-1)}{b^2-\zeta^2}\right)\right]\right\}$$

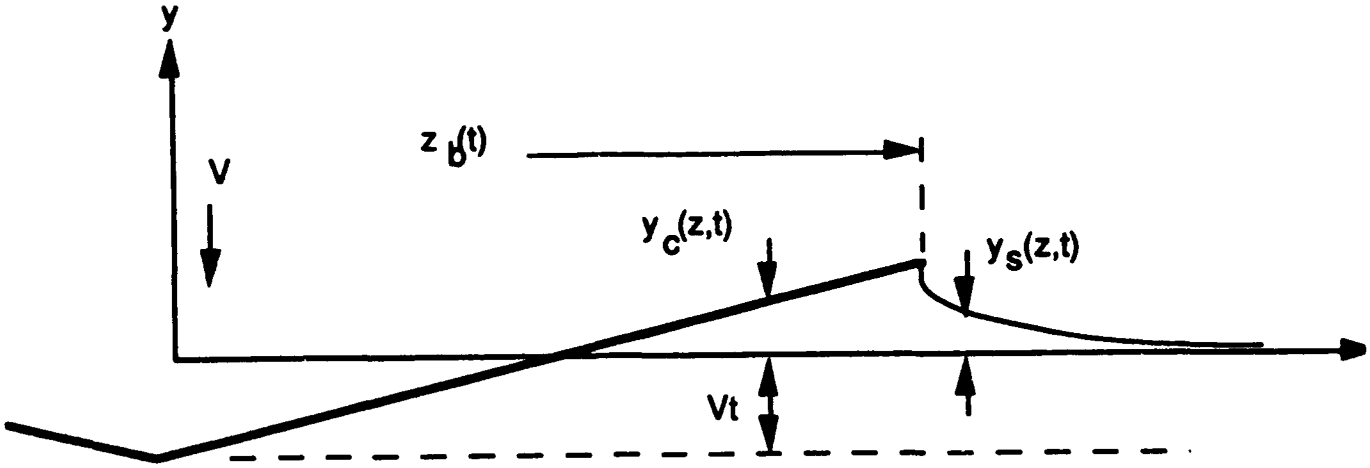
$$0 \le \zeta \le 1$$

Here, $\lambda \equiv \frac{1}{2} - \frac{\tilde{\beta}}{\pi}$ and F is the hypergeometric function of one argument with $\tilde{\beta} \equiv \tan^{-1}(\sin\beta)$.

VELOCITY CONTINUITY

Remove singular terms as:

$$1 + \frac{\gamma_s}{\pi}\frac{(b^2-1)^\lambda}{2\lambda}F\left(\lambda,\lambda,\lambda+1;1-b^2\right) = 0$$



$$\frac{\partial y_c(z,t)}{\partial t} = -1 = v(z,t) + \frac{1}{2}\gamma_c(z,t)\sin\beta$$

In terms of a displacement potential, ϕ^*:

$$y_c(\xi) = -1 + \xi z_{bt}\tan\beta = v^*(\xi) + \frac{1}{2}\gamma_c^*(\xi)\sin\beta$$

$$\frac{1}{2}\gamma_c^*(\xi)\sin\beta + \frac{1}{2\pi}\int_{\xi_0=-1}^{1}\frac{\gamma_c^*(\xi_0)}{\xi_0-\xi}d\xi_0 = -1 + \xi z_{bt}\tan\beta \qquad 0 \le \xi \le 1$$

$$\gamma_c^*(\xi) = -\frac{2\xi\cos\tilde{\beta}}{\sqrt{1-\xi^2}}\left(\frac{1-\xi^2}{\xi^2}\right)^{\frac{\tilde{\beta}}{\pi}}\cdot$$

$$\cdot\left\{1 - \frac{z_{bt}\cos\tilde{\beta}\tan\beta}{\sqrt{\pi^3}}\Gamma(\lambda)\Gamma(\frac{3}{2}-\lambda)\left[2 - \frac{1-\xi^2}{1-\lambda}F\left(\frac{1}{2},1,2-\lambda;1-\xi^2\right)\right]\right\} \qquad (30)$$

For continuous displacement at $\xi = z/z_b = 1$:

$$1 - \frac{2z_{bt}\cos\tilde{\beta}\tan\beta}{\sqrt{\pi^3}}\Gamma(\lambda)\Gamma(\frac{3}{2}-\lambda) = 0$$

UNKNOWNS: CONDITIONS:

Continuity of Displacement

zb,

$$z_{bt} = \frac{\sqrt{\pi^3}}{2\Gamma(\lambda)\Gamma(\frac{3}{2}-\lambda)\cos\bar{\beta}\tan\beta} \quad ; \lambda \equiv \frac{1}{2} - \frac{\bar{\beta}}{\pi} \ , \qquad \bar{\beta} \equiv \tan^{-1}(\sin\beta)$$

Continuity of Pressure

Vj,

$$V_j = z_{bt} + \sqrt{z_{bt}^2 + 1}$$

Continuity of Velocity

b (b = zb/zc)

$$1 + \frac{\gamma_s}{\pi}\frac{(b^2-1)^{\lambda}}{2\lambda} F\left(\lambda, \lambda, \lambda+1; 1-b^2\right) = 0 \qquad \gamma_s = -2V_j.$$

Wedge Vortex Distribution

$$\gamma_c(\zeta) = \frac{\gamma_s \cos\bar{\beta}}{\pi\lambda} Q^{\lambda} F(\lambda, \lambda, \lambda+1; Q) \ , \qquad \text{with} \quad Q(\zeta; b) \equiv \frac{\zeta^2\left(b^2-1\right)}{b^2-\zeta^2}$$

Wedge Pressure Distribution

$$C_p(\zeta) = 1 - \frac{1}{4}\gamma_c^2(\zeta) - z_{ct}\left[\gamma_c(1)(b-1) + \int_{\zeta_0=\zeta}^{1} \gamma_c(\zeta_0)d\zeta_0 + \zeta\gamma_c(\zeta)\right] \ , \quad z_{ct} = z_{bt}/b.$$

COMPUTATIONS

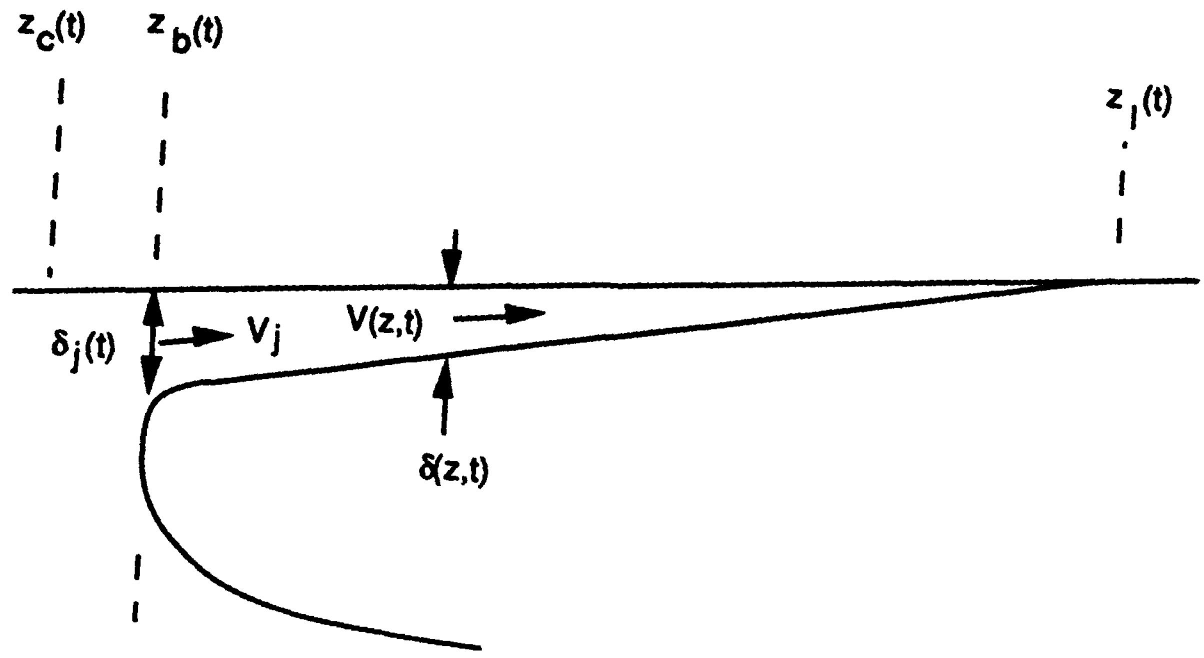
Wetting Factor ("jet rise")

$$WF = \frac{z_c(t)\tan\beta}{Vt} = z_{ct}\tan\beta$$

$$WF = \frac{\pi}{2}J(\lambda) \qquad \text{with, } J(\lambda) \equiv \frac{\sqrt{\pi}}{b\Gamma(\lambda)\Gamma(\frac{3}{2}-\lambda)\cos\tilde{\beta}} \text{ , and } \quad \lambda = \frac{1}{2} - \frac{\tilde{\beta}}{\pi}$$

WEDGE SOLUTION CHARACTERISTICS
TABLE II

β, degrees	V_j	b	WF	J	C_f
5	34.63	1.00063	1.512	.9629	846.3
10	16.65	1.00192	1.460	.9293	180.3
15	10.68	1.00340	1.413	.8997	68.93
20	7.711	1.00491	1.373	.8740	33.56
25	5.944	1.00640	1.338	.8519	18.68
30	4.777	1.00790	1.308	.8328	11.32



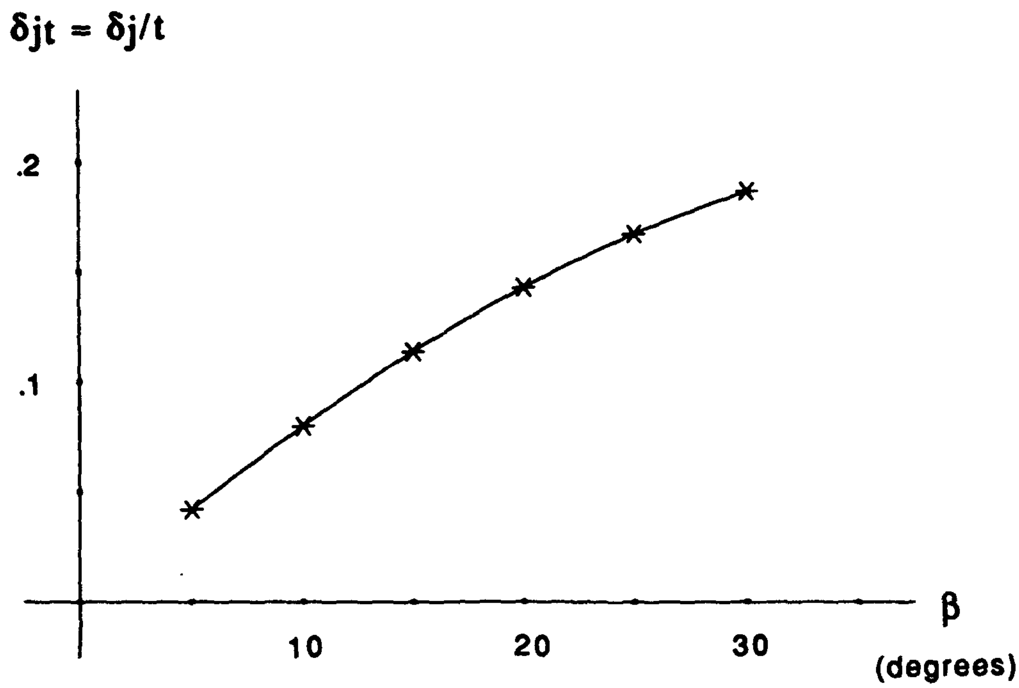
TRUNCATED JET CHARACTERISTICS
FIGURE 7

For continuity of mass, to second order:

$$\delta_{jt} = -\frac{2z_{bt}}{V_j}\left(\frac{1}{2}z_{bt}\tan\beta + A_s - 1\right)$$

From the displacement potential:

$$A_s = \frac{1}{2\pi}\frac{\cos\bar{\beta}}{1-\lambda}\int_{\xi_0=0}^{1} \xi_0^{2\lambda}\ln\left(\frac{1+\xi_0}{1-\xi_0}\right)\left(1-\xi_0^2\right)^{1-\lambda} F\left(\frac{1}{2},1,2-\lambda;1-\xi_0^2\right)d\xi_0$$



JET THICKNESS AT JET-HEAD TRUNCATION
FIGURE 8

Jet thickness distribution, for $C_p = 0$:

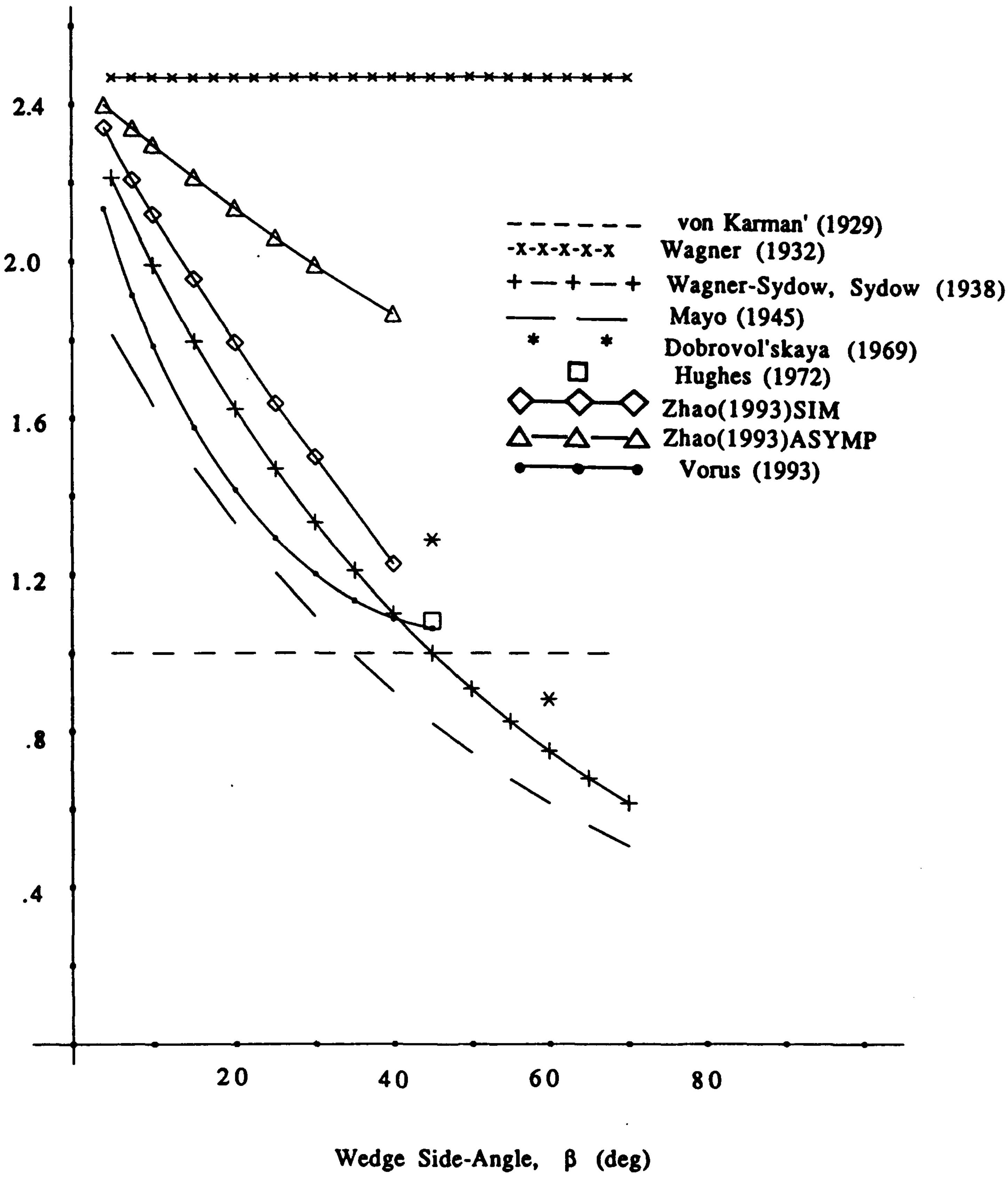
$$\delta(s) = \delta_j \left(\frac{V_j - s}{V_j - z_{bt}} \right) \qquad z_{bt} \leq s \leq z_{lt} \tag{56}$$

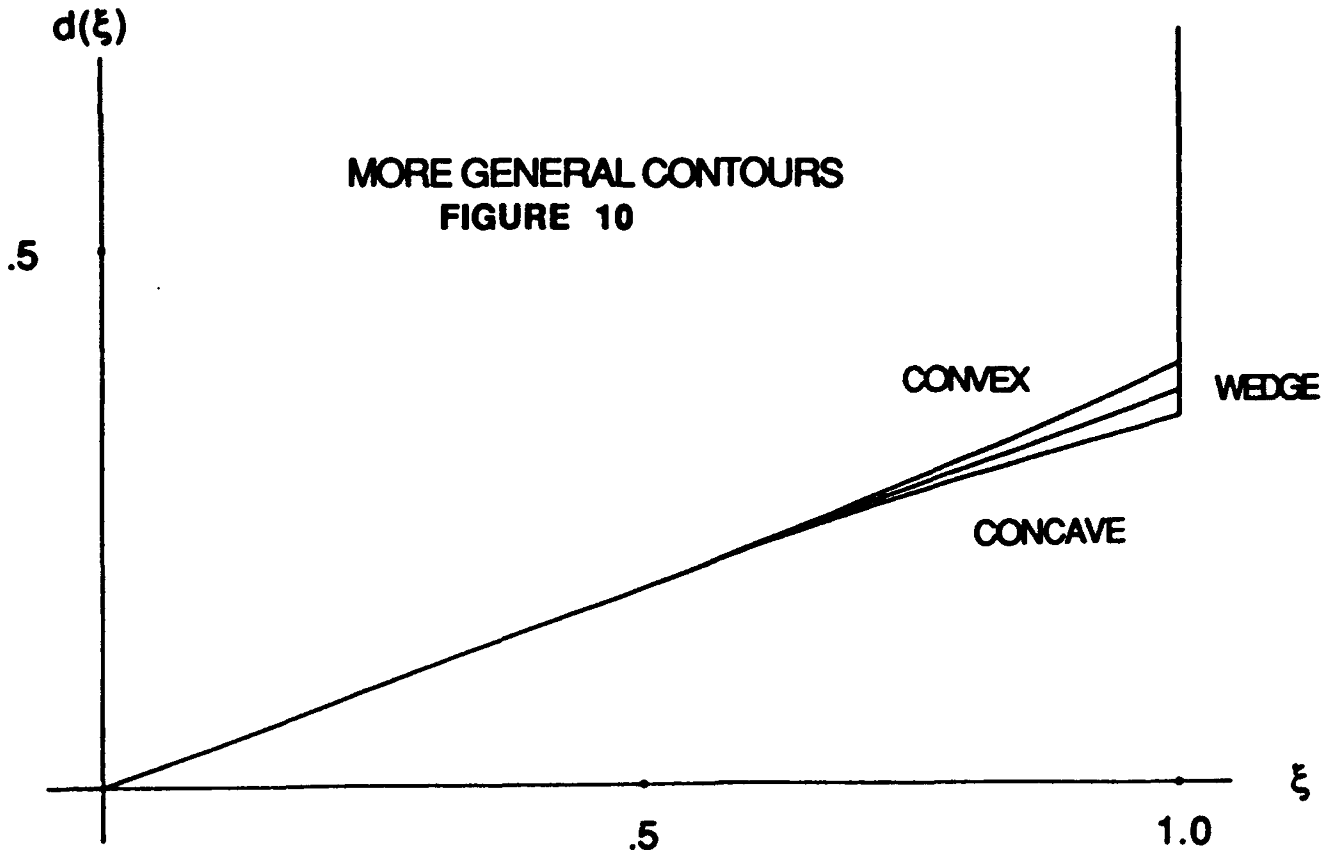
For $\delta(z_{lt}) = 0$:

$$z_{lt} = V_j \tag{57}$$

$$m' = \frac{m}{\frac{1}{2}\rho\pi\left(Y_{wl}\tan\beta\right)^2} = \frac{1}{\pi}C_f\tan^2\beta$$

ADDED MASS OF IMPACTING WEDGES
FIGURE 6



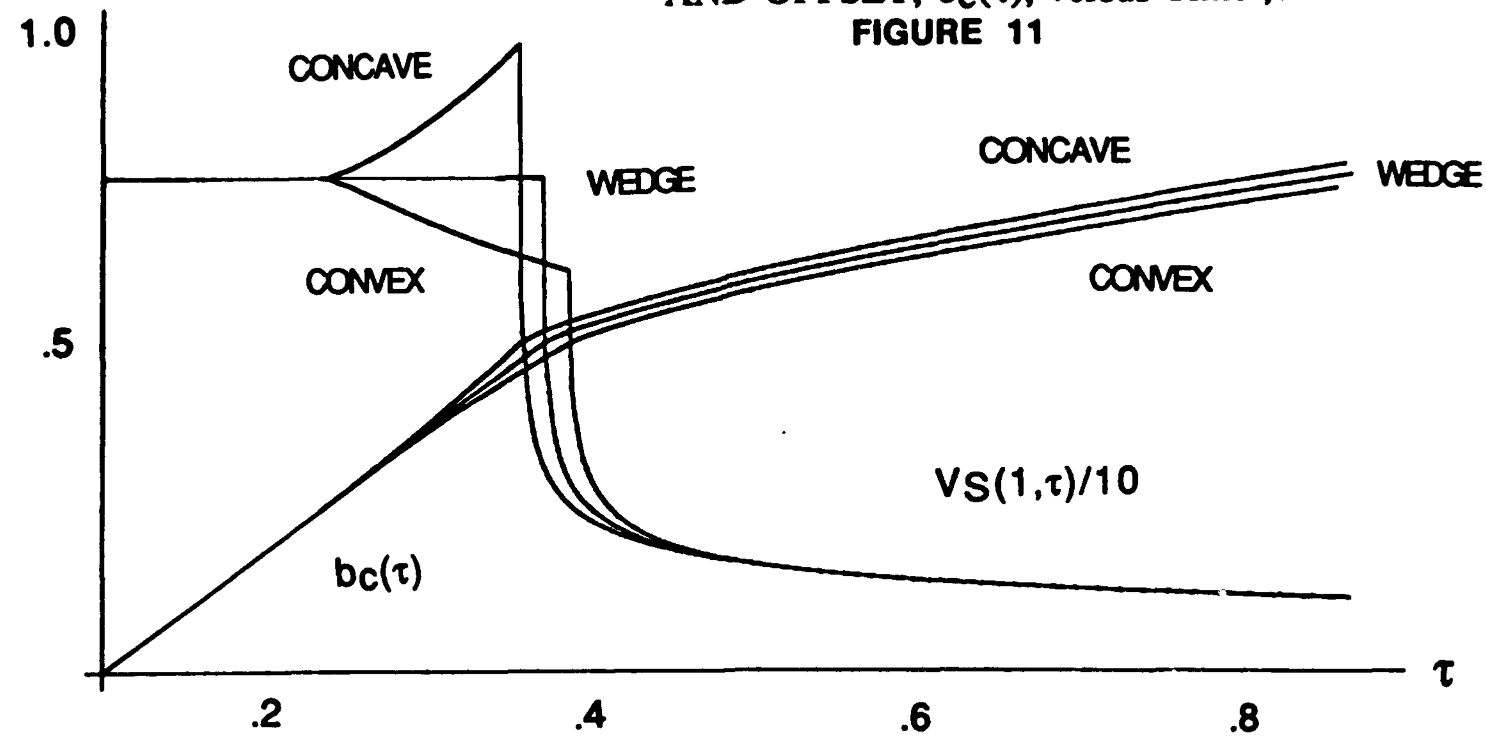


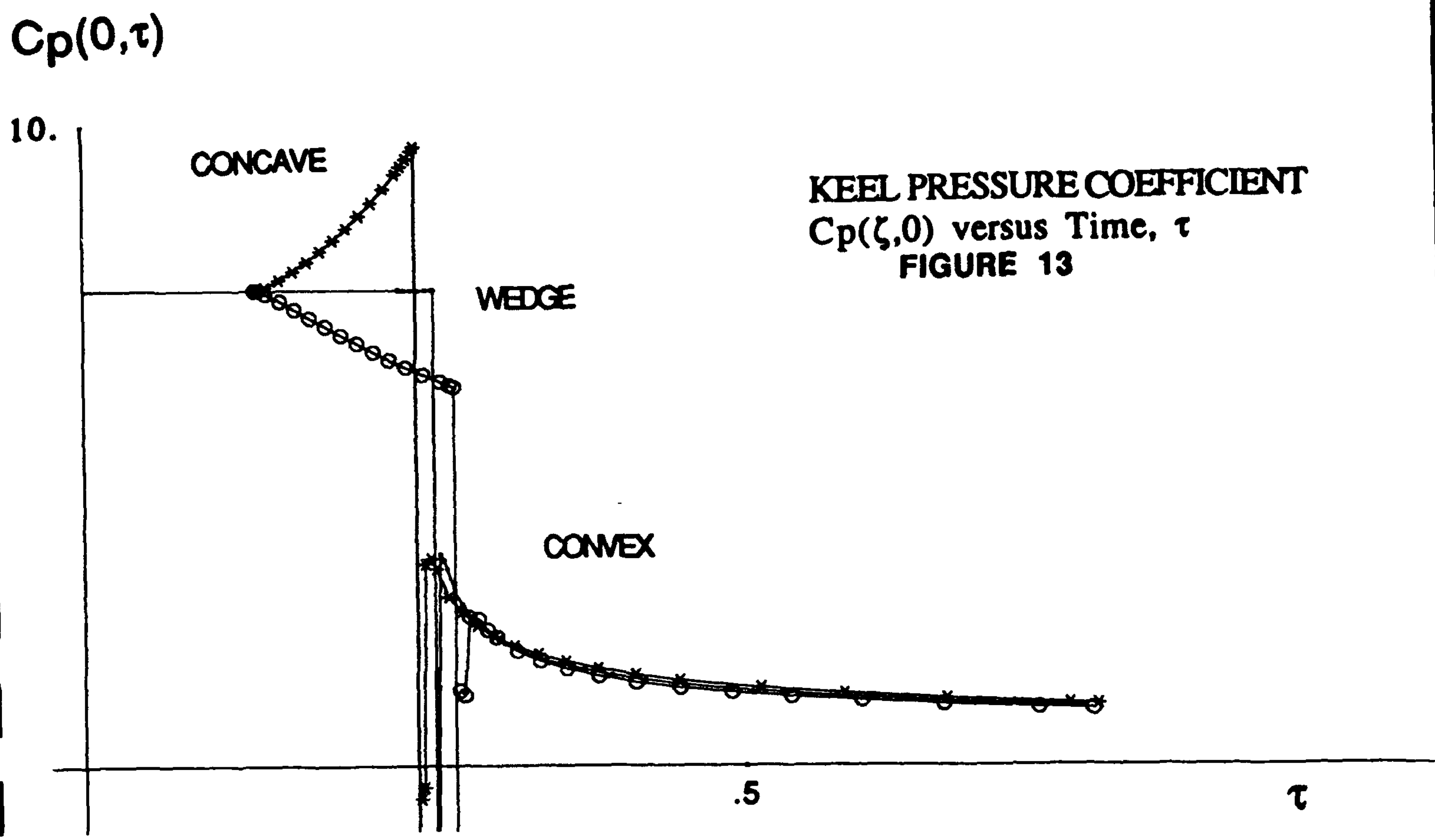
MORE GENERAL CONTOURS

FIGURE 10

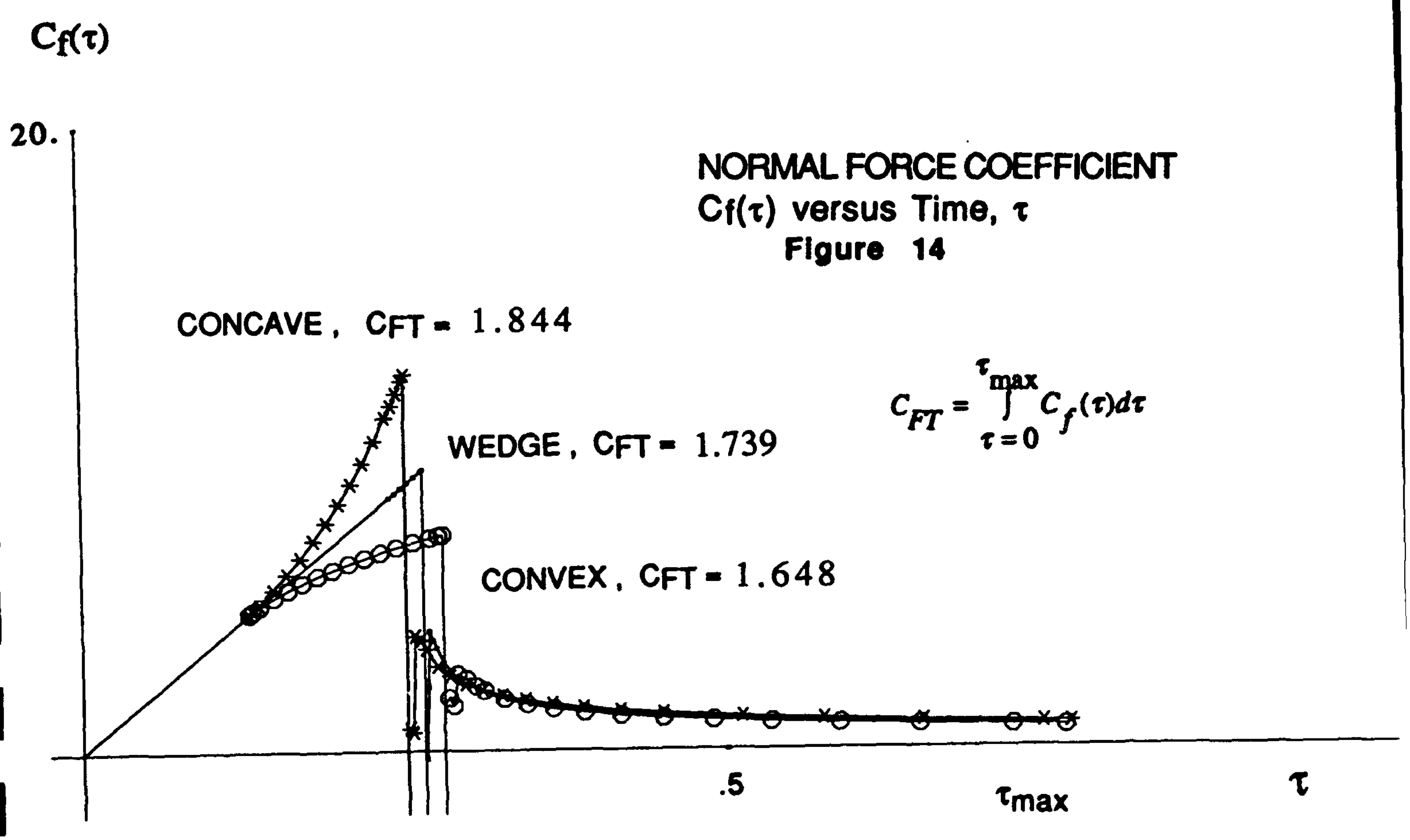
JET-HEAD SEPARATION VELOCITY, Vs(1,τ), AND OFFSET, $b_C(\tau)$, versus Time ,τ

FIGURE 11



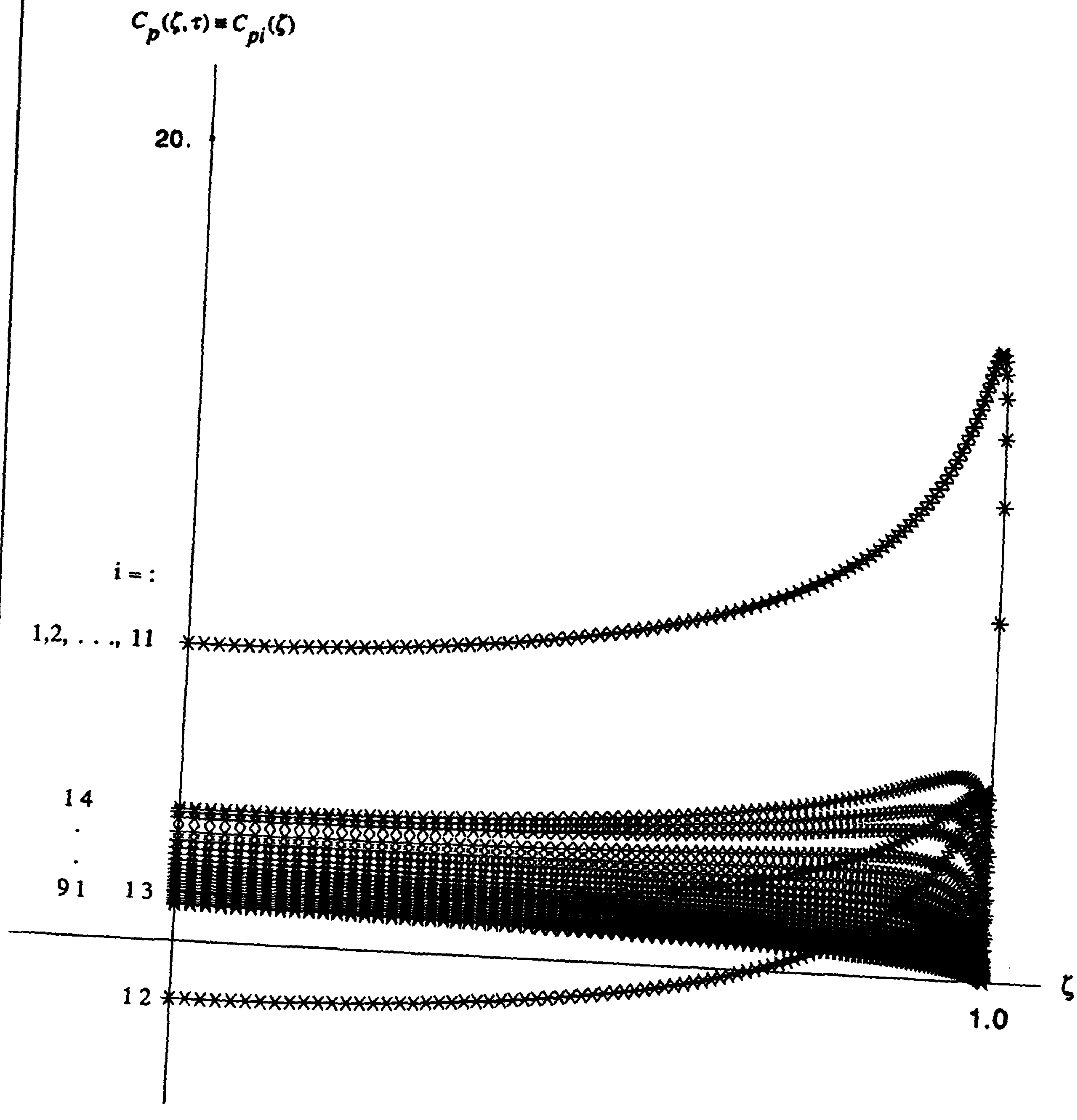


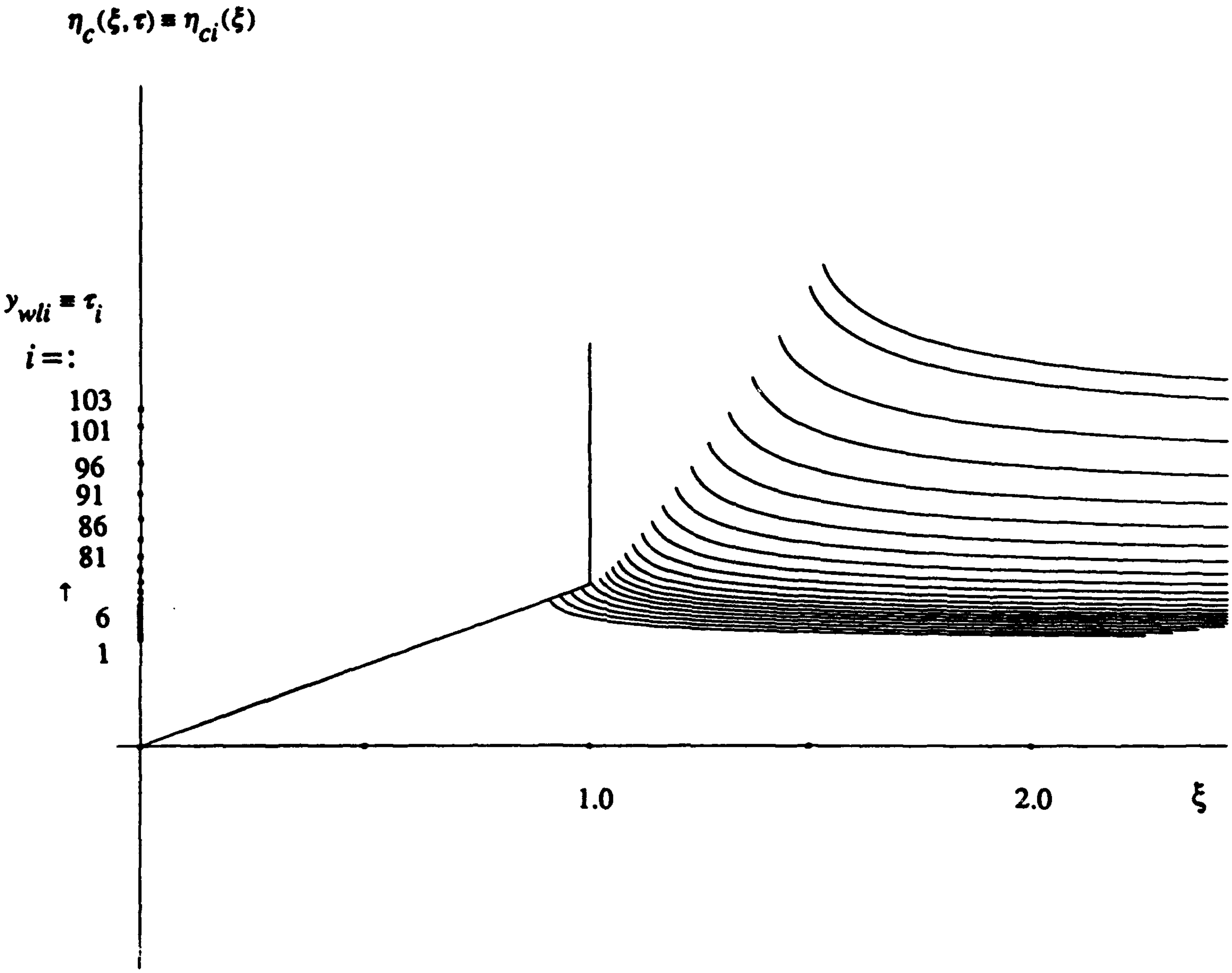
KEEL PRESSURE COEFFICIENT
Cp(ζ,0) versus Time, τ
FIGURE 13



NORMAL FORCE COEFFICIENT
Cf(τ) versus Time, τ
Figure 14

WEDGE CONTOUR PRESSURE DISTRIBUTION
FIGURE 15





FREE-SURFACE DISPLACEMENT
WEDGE CONTOUR
FIGURE 12

FUTURE WORK

Nonsymmetric Cylinders

Linear Gravity Flow Beyond the Jet Head

Lifting Surface (3D) Corrections

Extraction (inverse impact)

Rapid Lateral Expansion of Thin Cylinders (vertical line-source)

Gas Entrapment

ONR WORKSHOP

NONLINEAR SEA LOADS AND SHIP RESPONSE: A BASIS FOR SHIP STRUCTURAL DESIGN

Nonlinear Hydrodynamic Forces on High Speed Vessels

presented by

Armin W. Troesch, PhD, PE

Department of Naval Architecture and Marine Engineering

The University of Michigan

Ann Arbor, Michigan

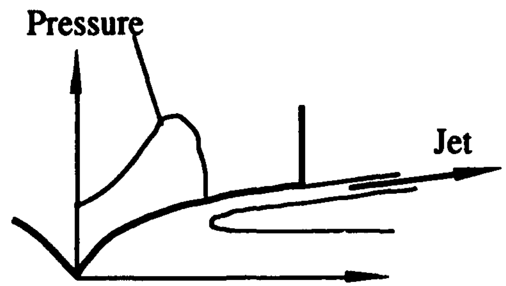
Issues Related to the Hydrodynamics of Planing Hull Seakeeping

Physical Modeling

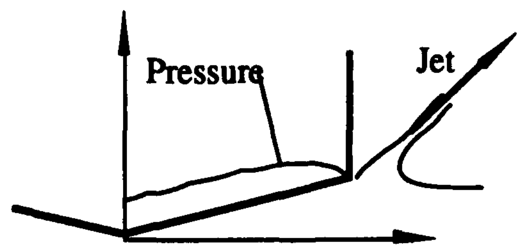
- Boundary Conditions
- Determination of Wetted Surface
- Chines Dry or Chines Wet Flow
- Importance of Jet Hydrodynamics

Validation

- Two-dimensional Theories
- Constant Forward Speed Limit
- Experiments



a) Schematic of chines unwetted condition with pressure distribution, forward station.



b) Schematic of chines wetted condition with pressure distribution, after station.

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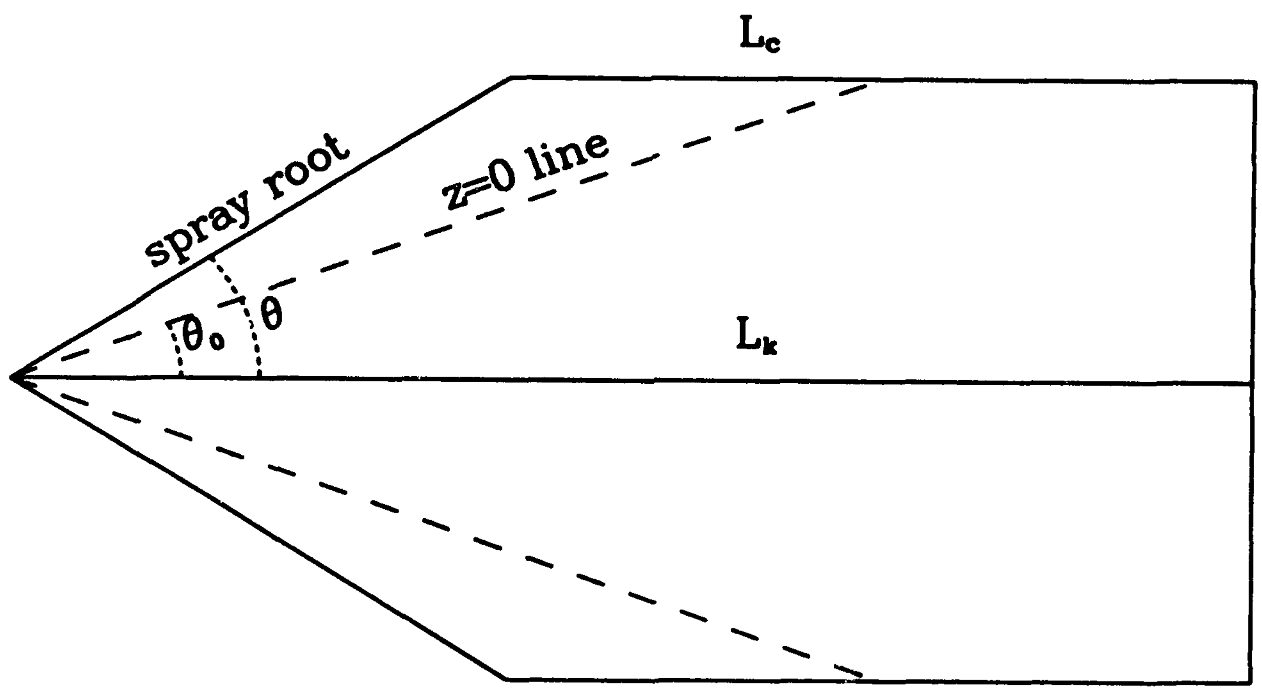
Wetted surface platform



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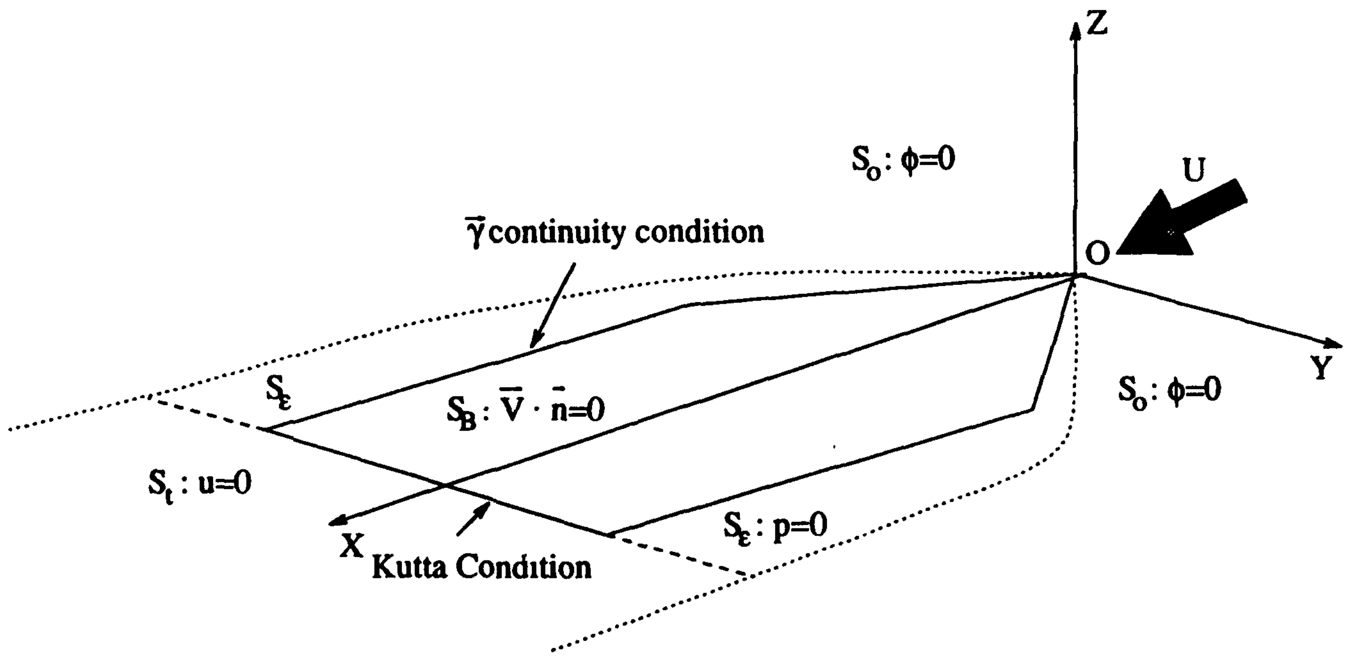
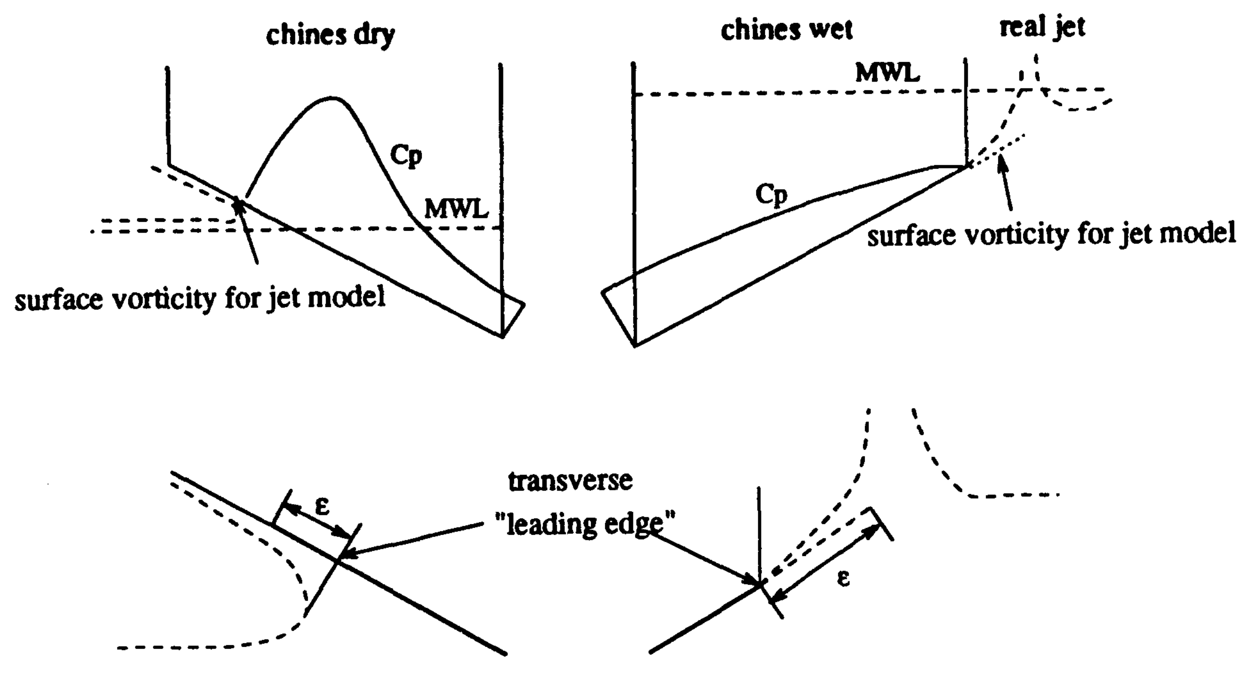


Schematic of computational domain for the z=0 model



Jet location and modeling details for both chines dry and wet parts

for the z=h model

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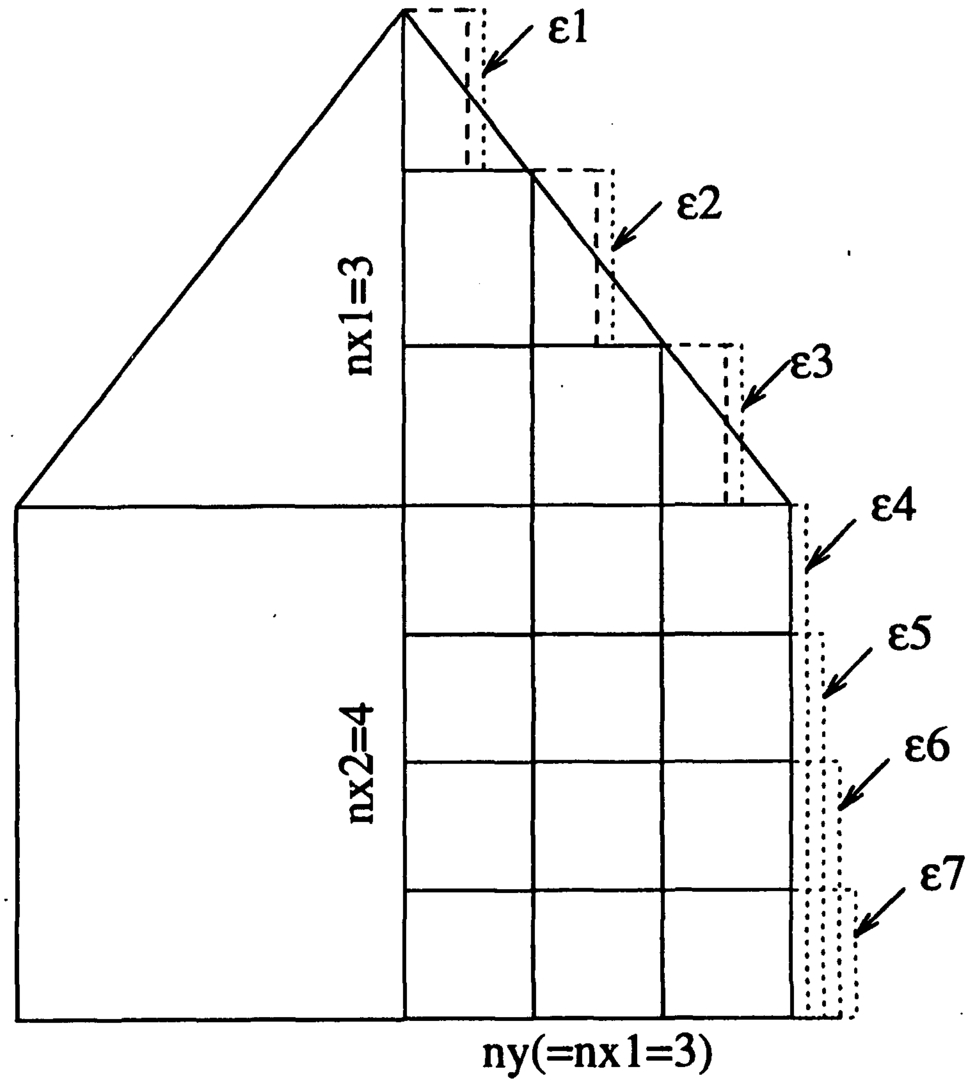

Panel number: nx1*(nx1+1)/2+nx1*nx2

Rectangular panels: panel1(nx1,nx2)

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Classification of models

	I	II	III	IV
linearity	linear	nonlinear		
location	$z = 0$	$z = h$		
wetted surface	B, M	Z and F	V(l)	V(nl)
jet approach	$\gamma_j(x, y) = \gamma_j(x)$	γ_j convects		
panel shape	rectangular	trapezoidal		
body type	chines dry	chines dry and wet		

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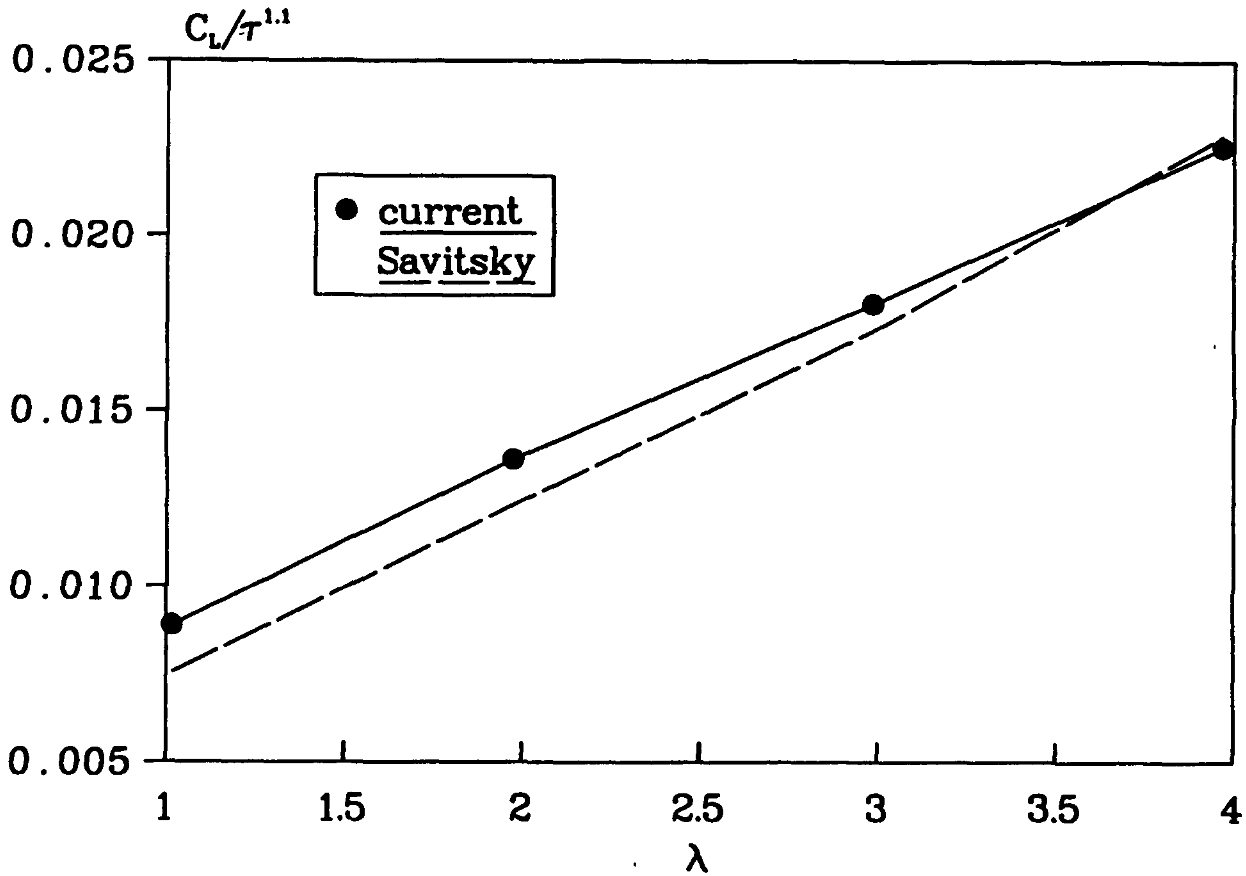
Validation of numerical model



$\beta = 20°$, $\tau = 5°$, $\theta = 20.4°$, and $F_n = 5$.

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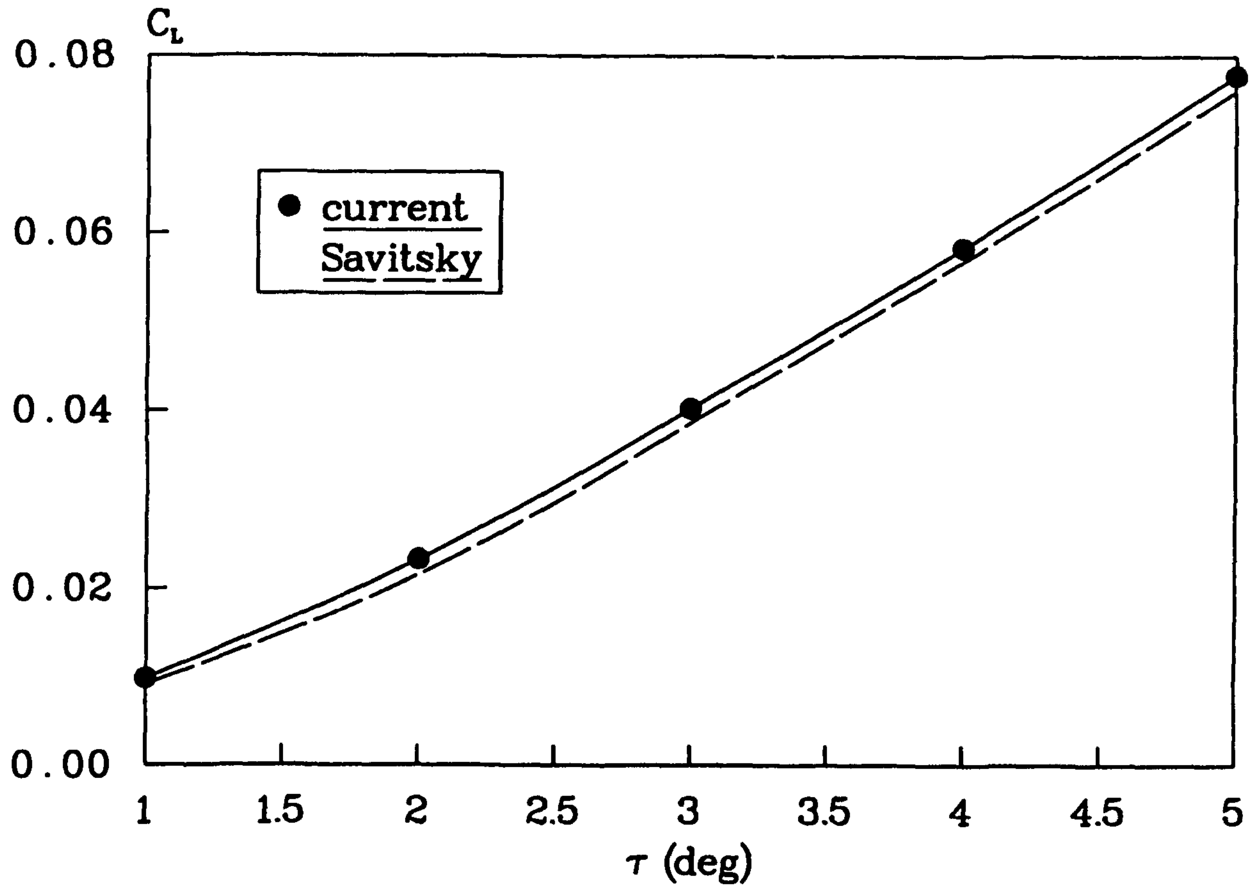
Validation of numerical model



$\beta = 20°$, $\lambda = 2.5$, and $F_n = \infty$.

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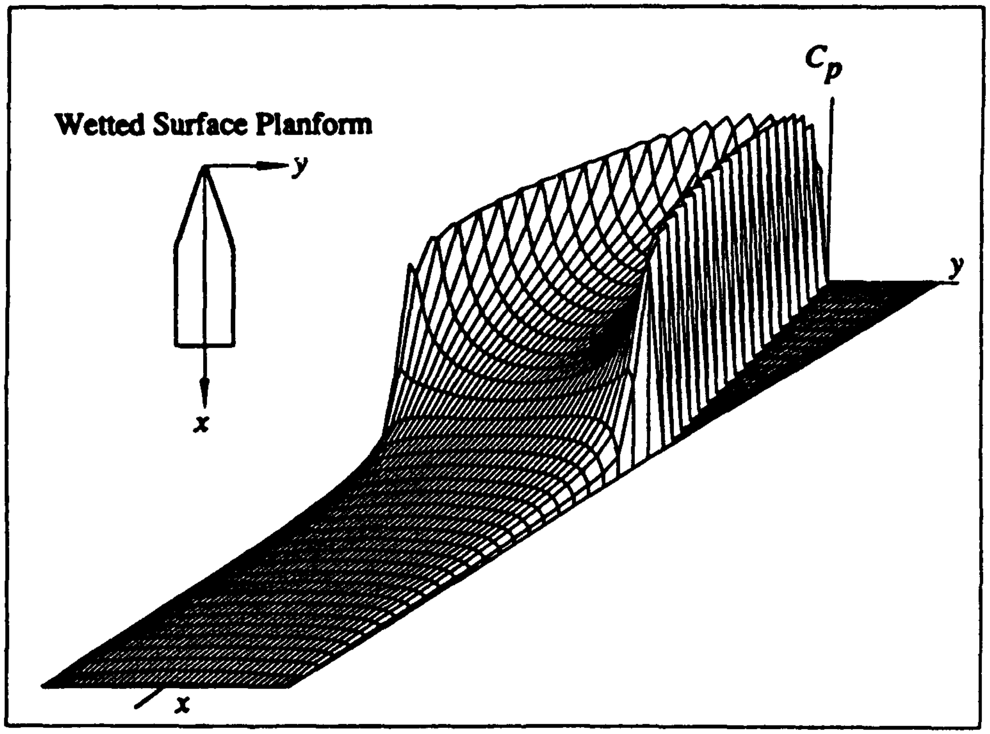

Cp calculated from a vortex lattice method.
Deadrise: 20 degrees; Trim: 4 degrees;
Mean wetted length/beam ratio: 2.56.

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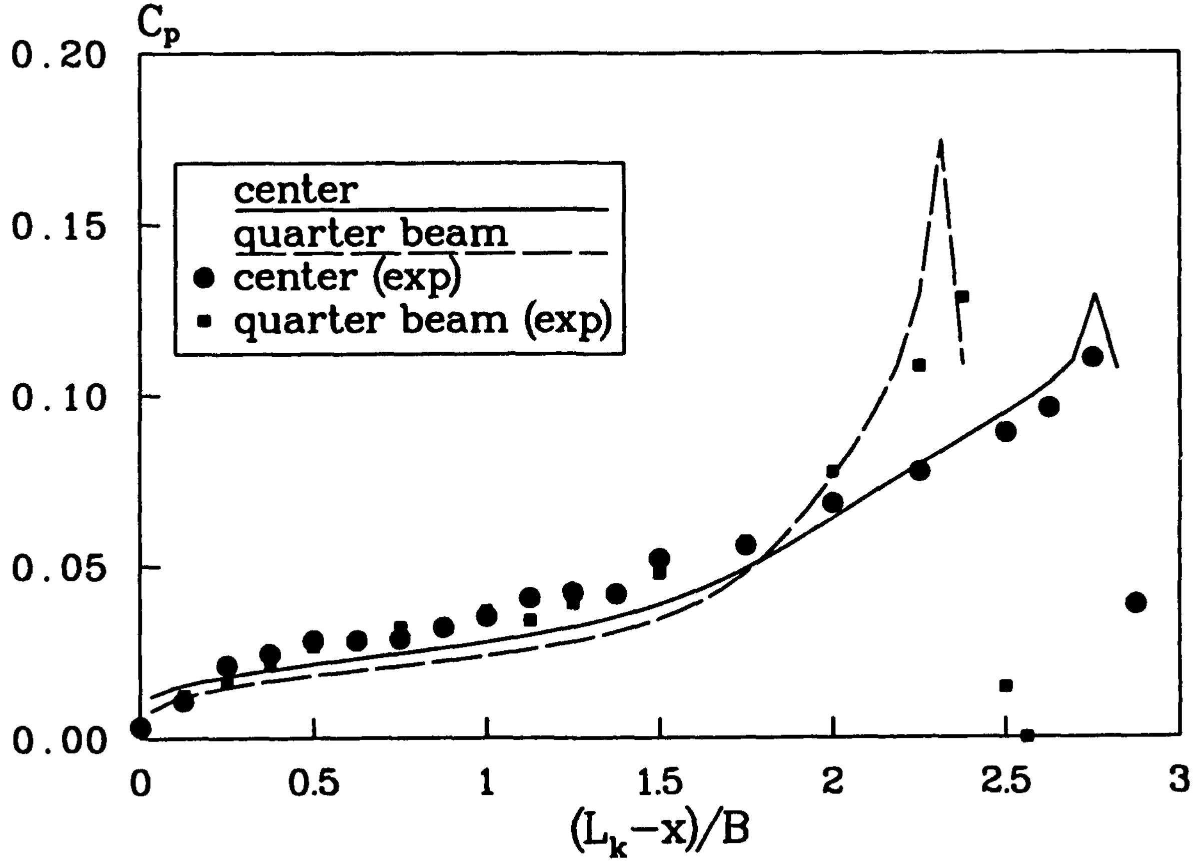
Validation of numerical model



The z=h model

$\beta = 20°$, $\tau = 6°$, $\theta = 27.7°$, $\lambda = 2.4$, and $F_n = 12$.

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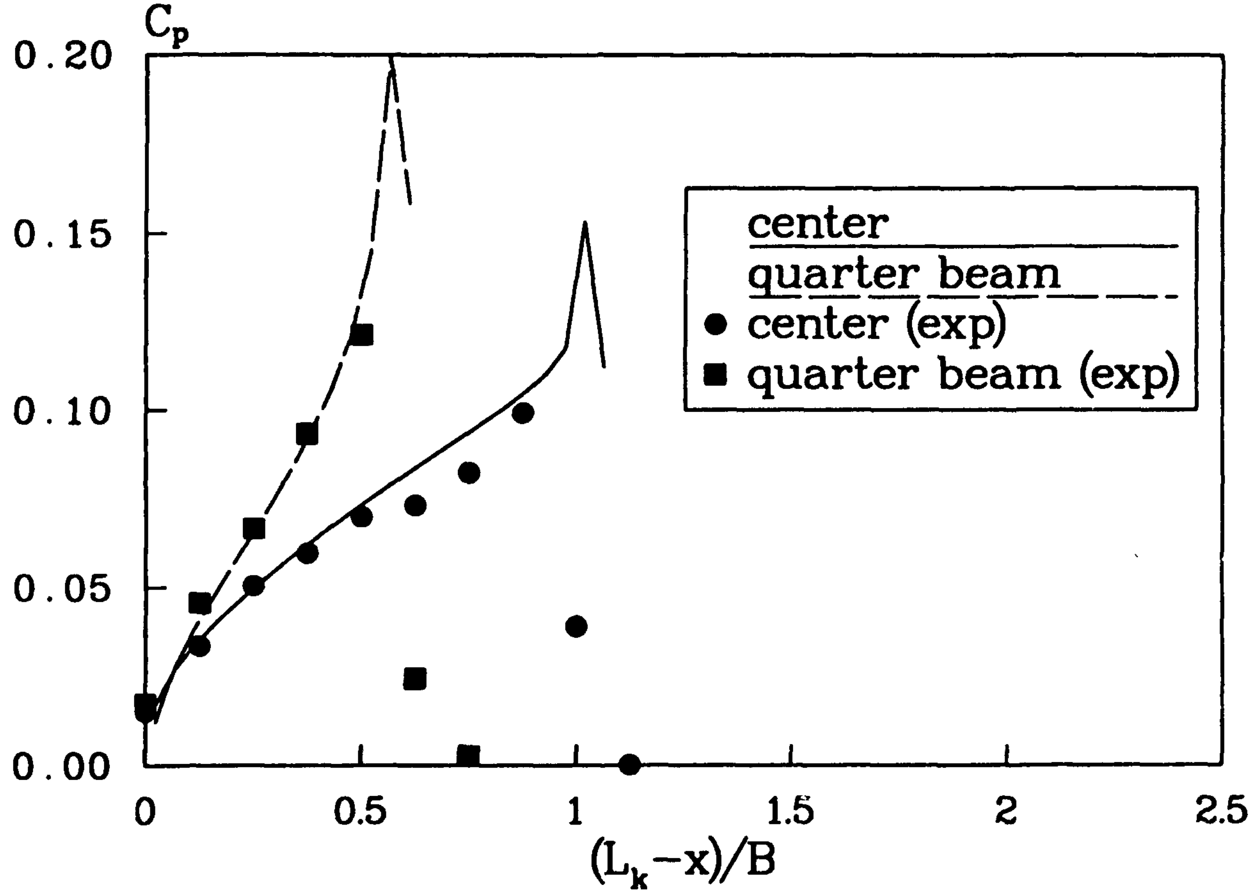
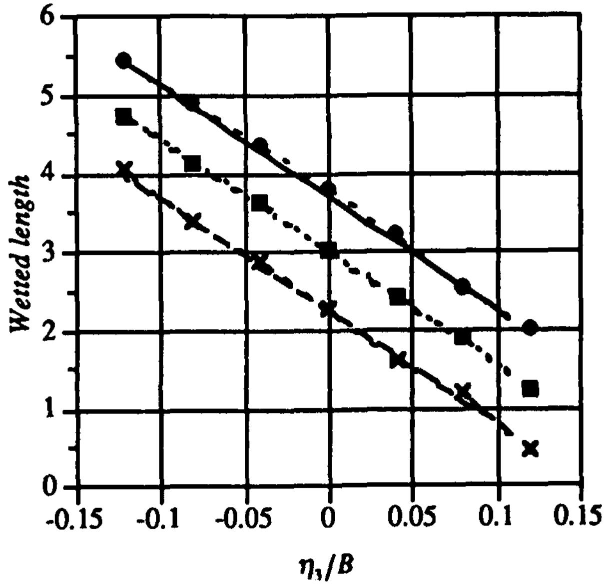
Validation of numerical model

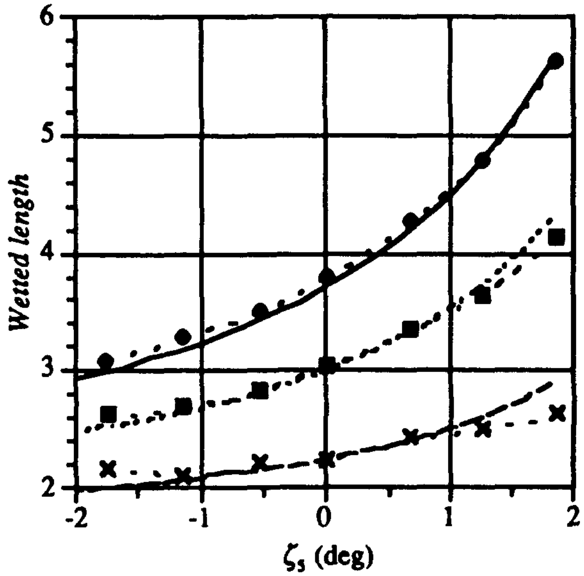


The z=h model

$\beta = 20°$, $\tau = 6°$, $\theta = 27.7°$, $\lambda = 0.611$, and $F_n = 12$.

MOTION DEPENDENT WETTED LENGTH





Wetted length versus constant heave displacement or constant pitch rotation. -■-, λave; - - - ; -●-, λk; —; -✖-, λc ; - - - .

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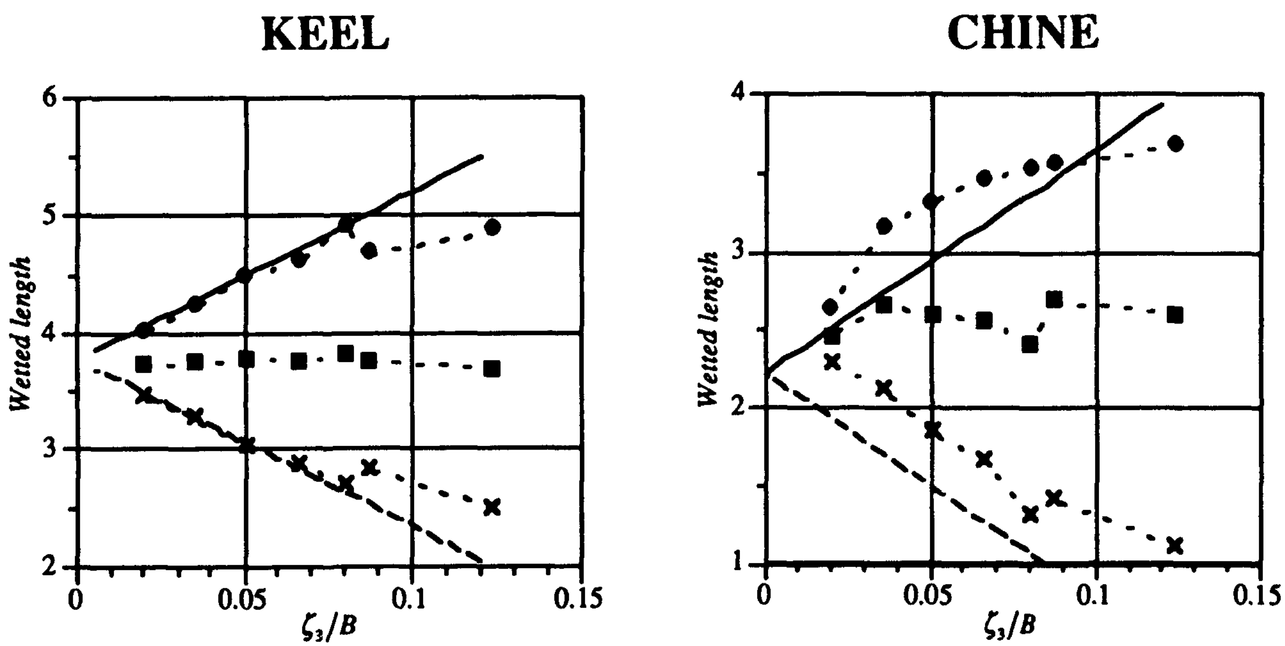
DYNAMIC BEHAVIOR OF WETTED LENGTH



Wetted maximum and minimum lengths versus heave amplitude.
$L/B = 3.0, Fn = 2.0, \tau = 4.0$ deg, $\underline{\omega} = 1.13$

-■-, λave; - - - ; -●-, λmax; —; -✖-, λmin ; - - - .

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Evaluation hydrodynamic loads

$$F_3(t) = -\int_{L(t)} d\xi \int_{B(\xi,t)} d\eta \, n_3(\xi,\eta,t)\, p(\xi,\eta,t)$$

Linear analysis (heave) for validation of model

$$C_{3j} \equiv -\frac{\partial F_3}{\partial \zeta_j}$$
$$B_{3j} \equiv -\frac{\partial F_3}{\partial \dot{\zeta}_j}$$
$$A_{3j} \equiv -\frac{\partial F_3}{\partial \ddot{\zeta}_j}$$

where derivatives are evaluated in the mean position of the hull.

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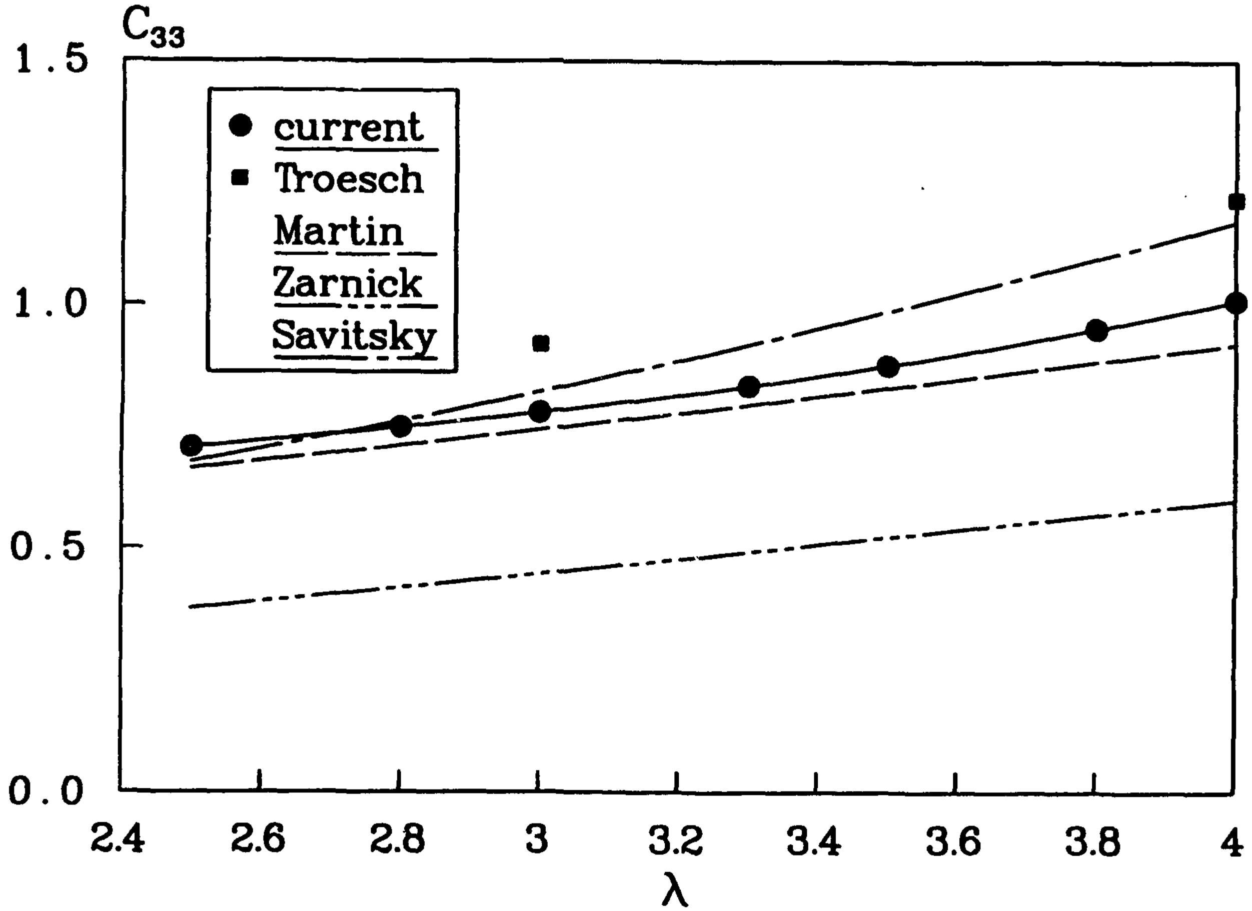
Force stiffness due to heave



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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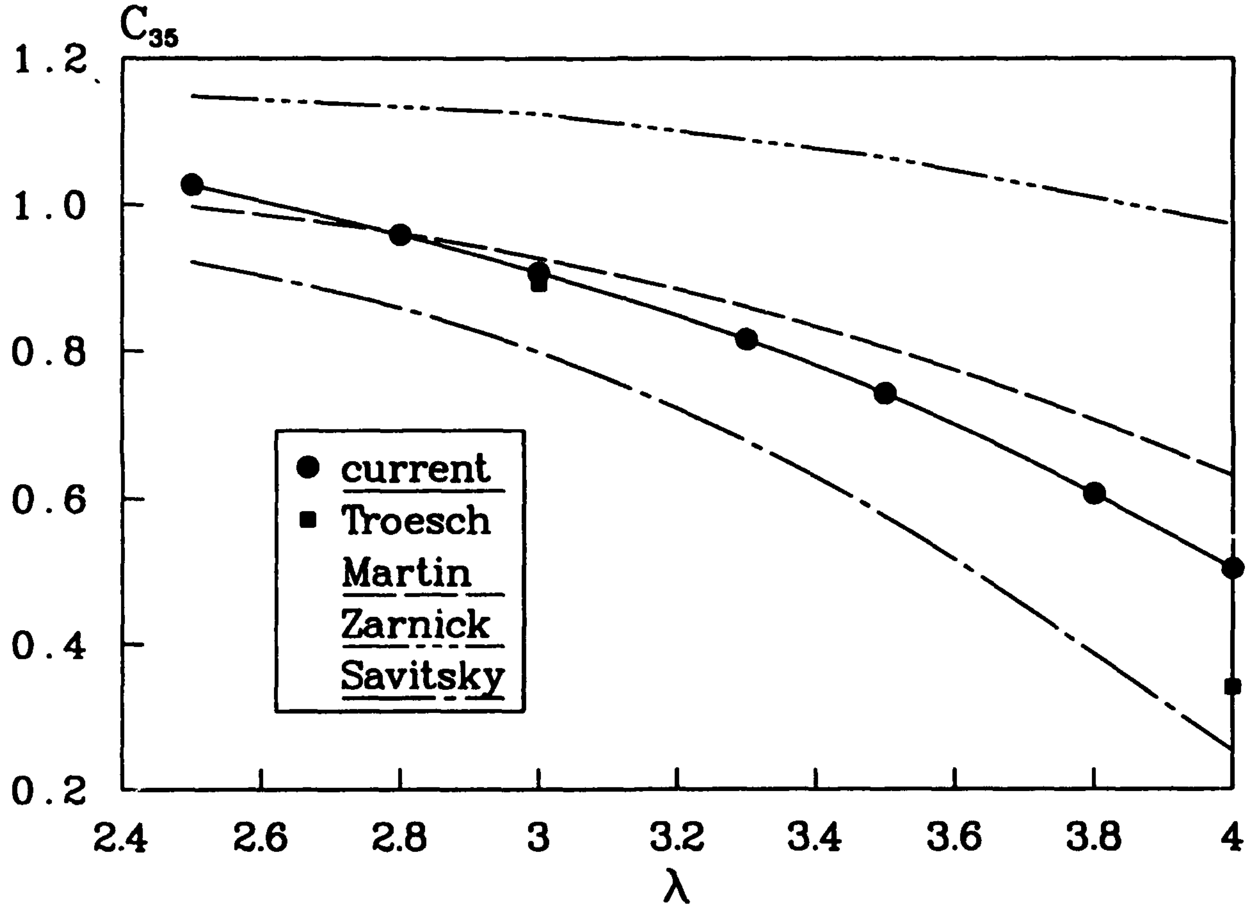
Force stiffness due to pitch



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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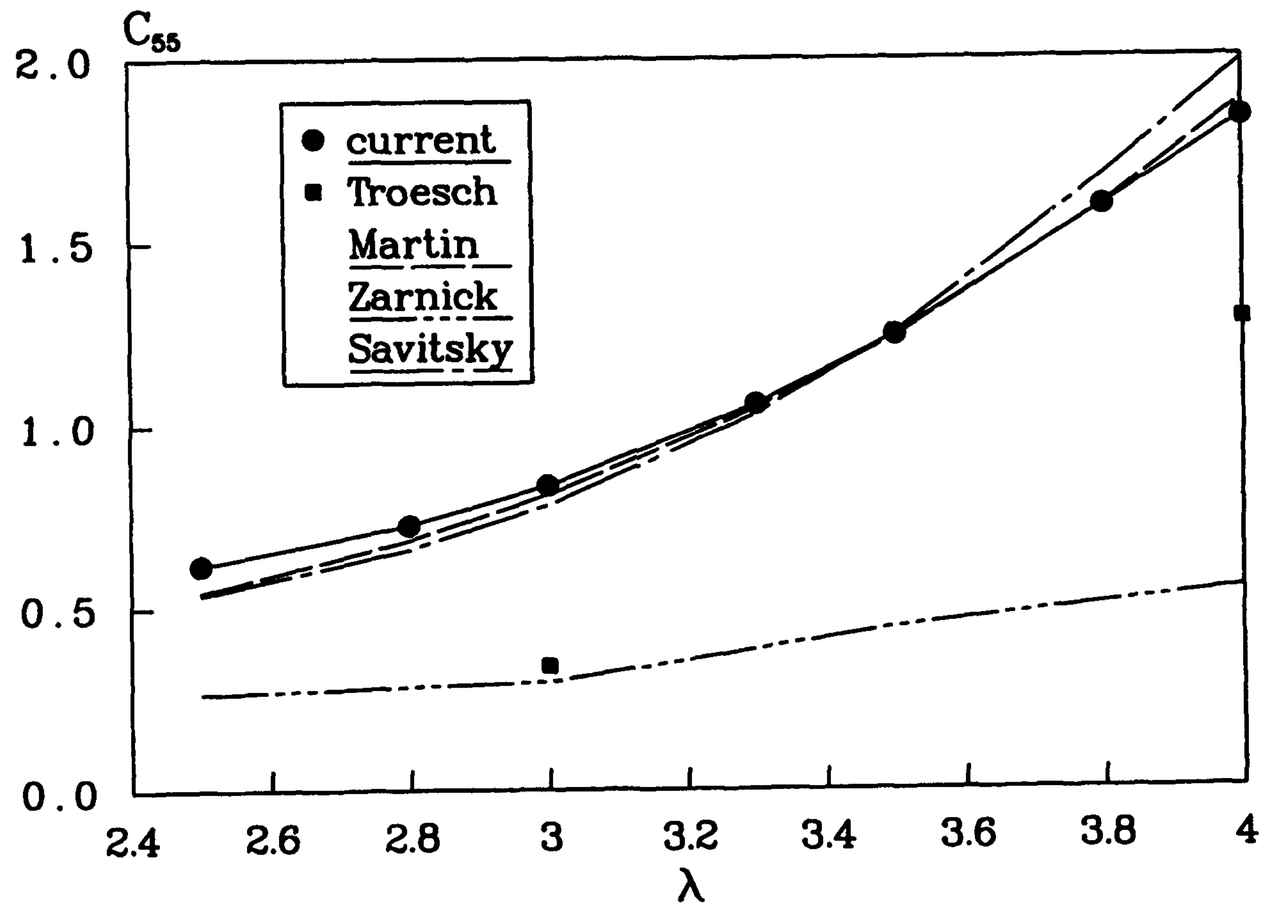
Moment stiffness due to pitch



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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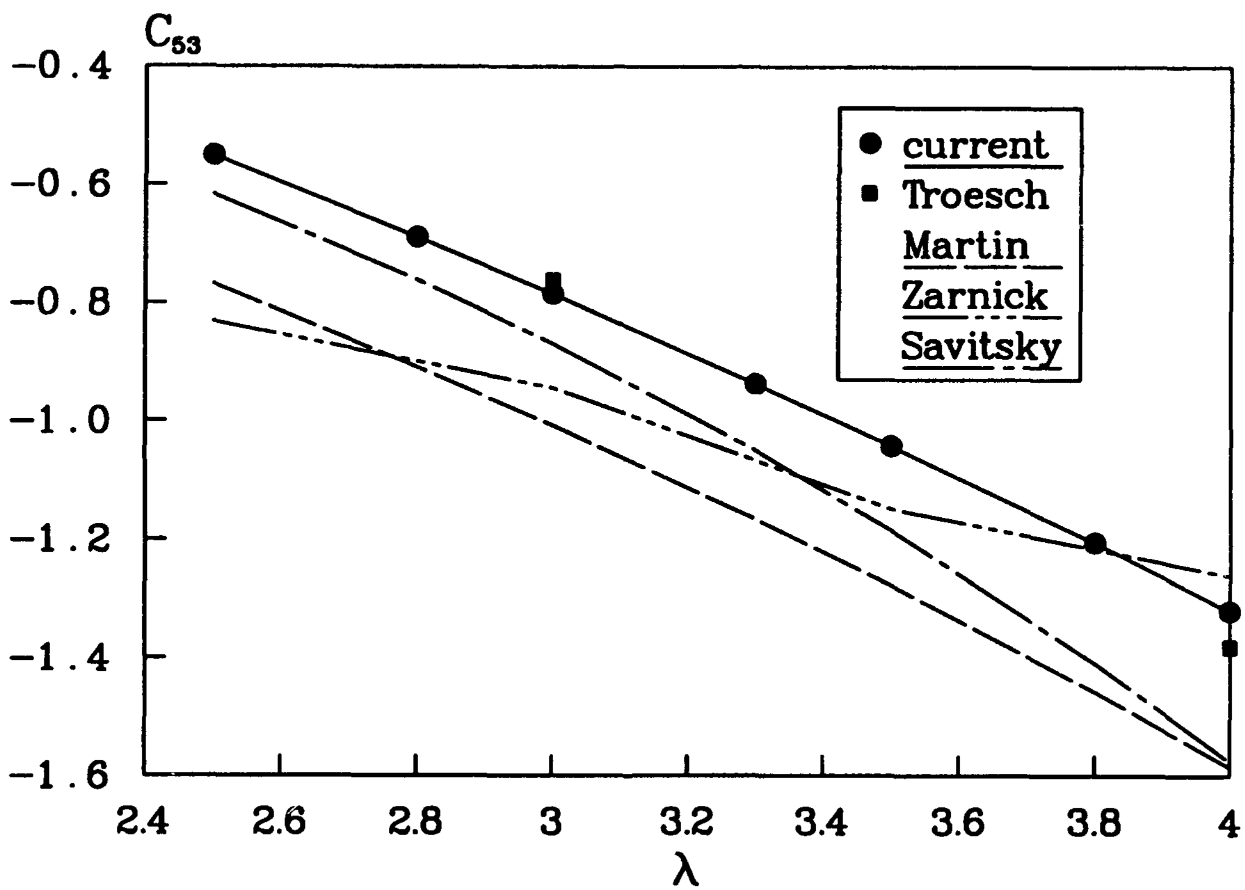
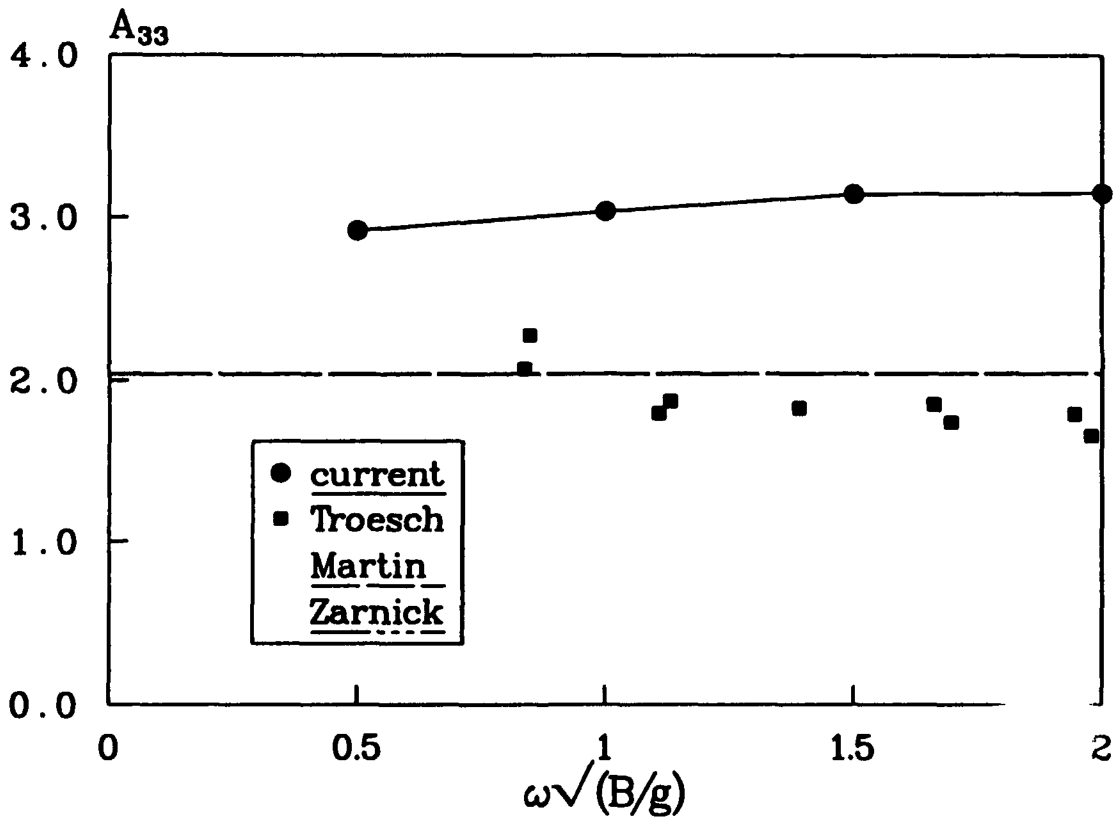
Moment stiffness due to heave



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

Hydrodynamic inertial force due to heave

(First order)



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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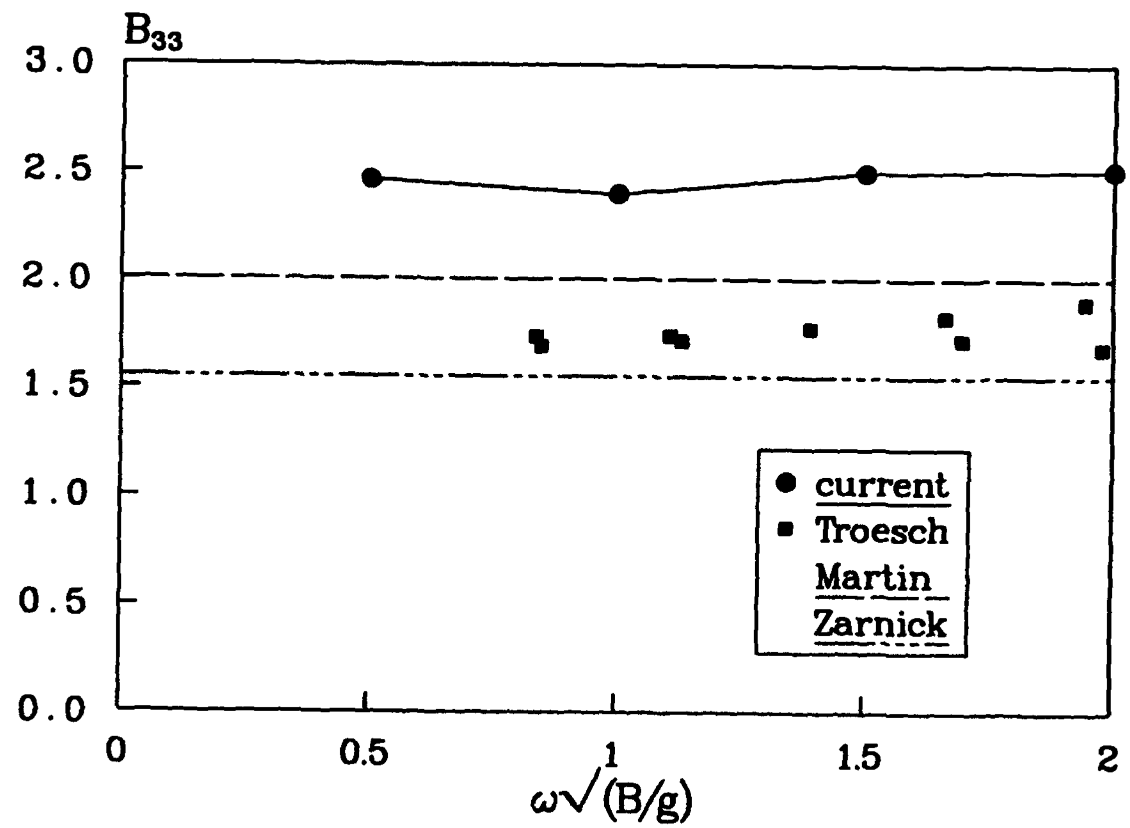
Hydrodynamic damping force due to heave

(First order)



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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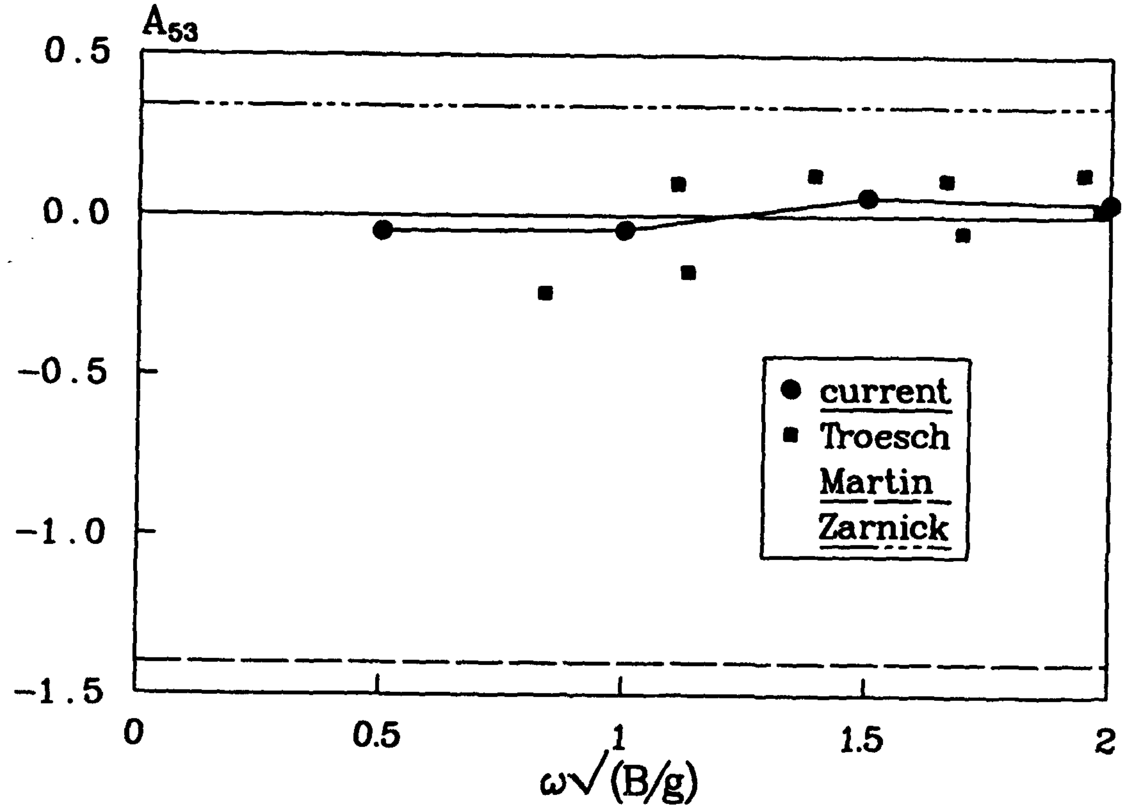
Hydrodynamic inertial moment due to heave

(First order)



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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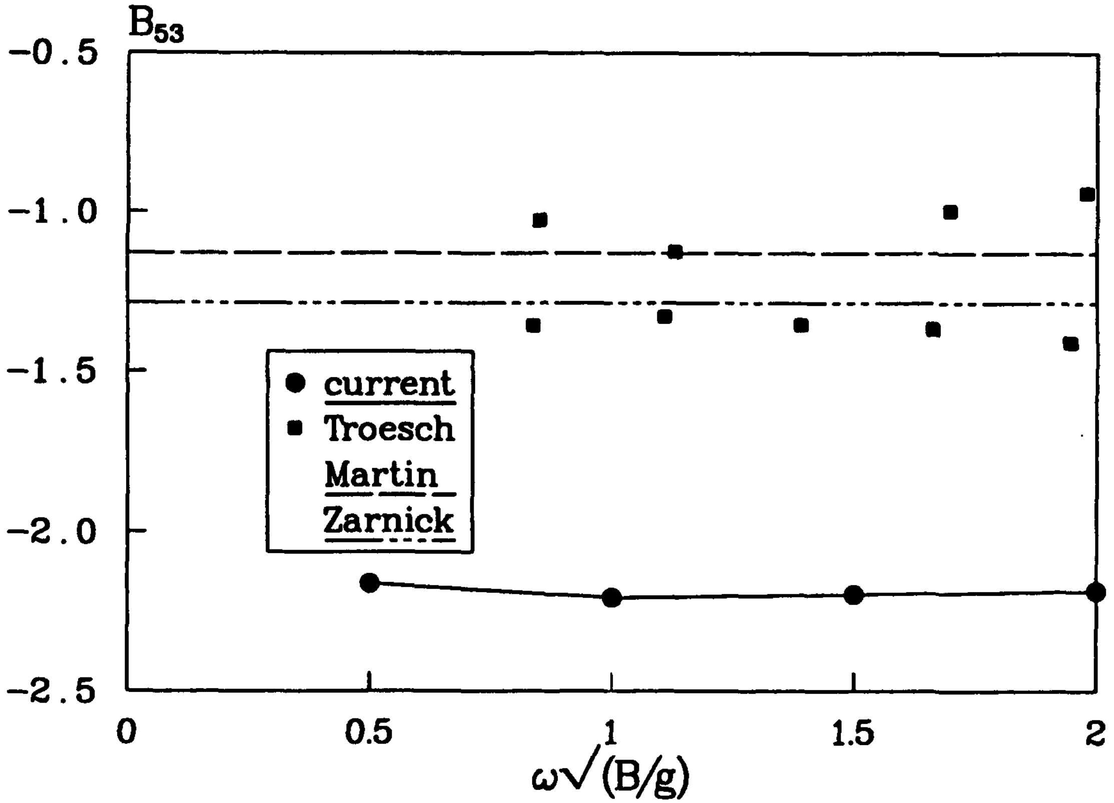
Hydrodynamic damping moment due to heave

(First order)



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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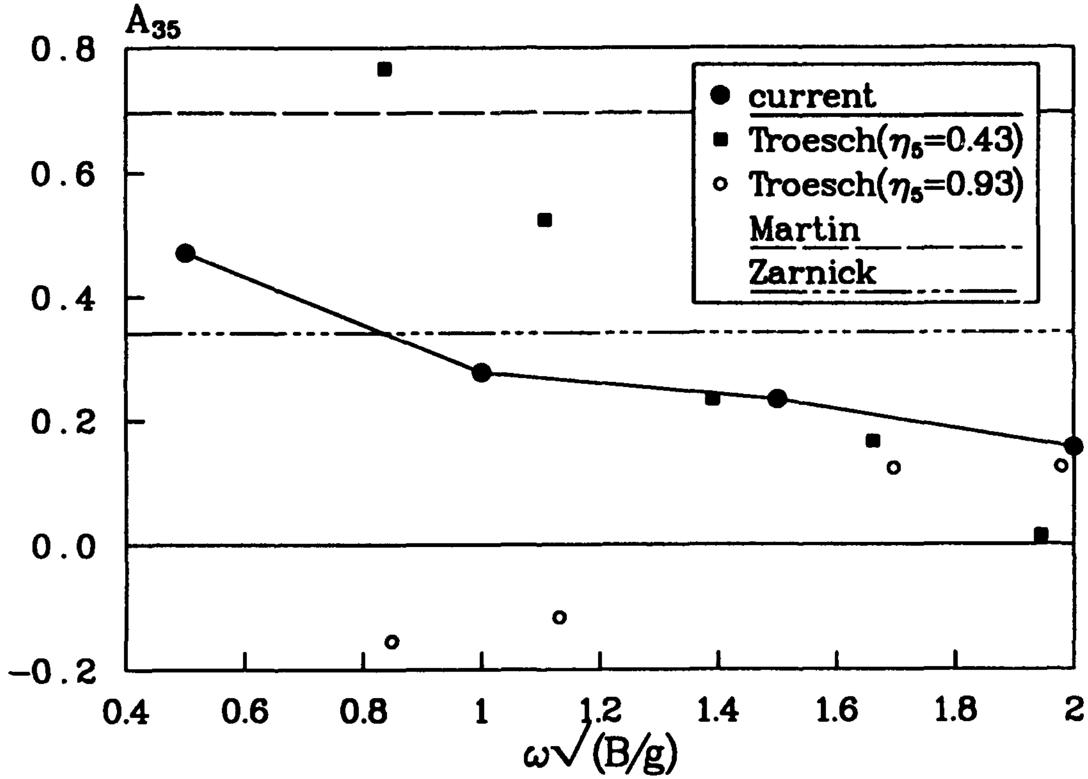
Hydrodynamic inertial force due to pitch

(First order)



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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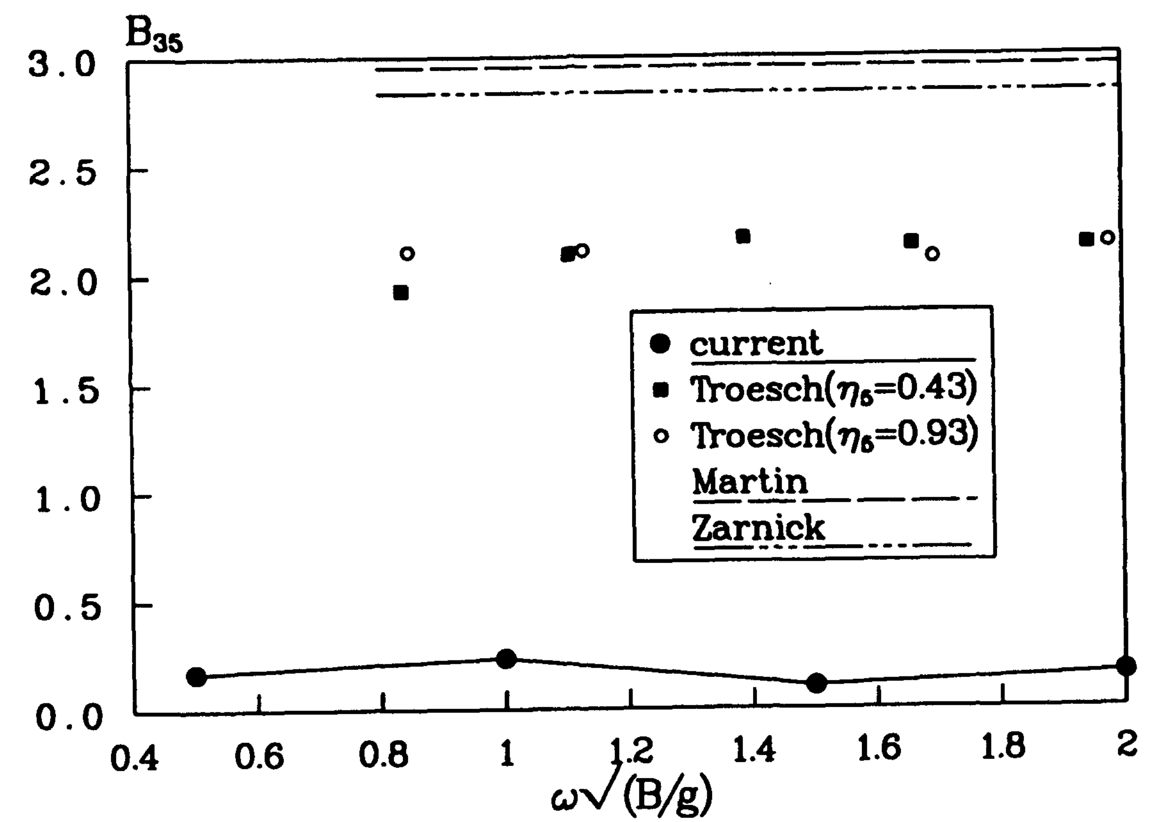
Hydrodynamic damping force due to pitch

(First order)



$\beta = 20°$, $\tau = 4°$, B=0.318m, $\lambda = 3$, $L_{cg}/B = 1.47$, $V_{cg}/B = 0.65$, and $F_n = 2.5$.

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SHIP STRUCTURES

Jerome Sikora

CDNSWC

CODE 66.1

Bethesda, MD 20084-5000

NSWC

Naval Surface Warfare Center

CARDEROCK DIVISION

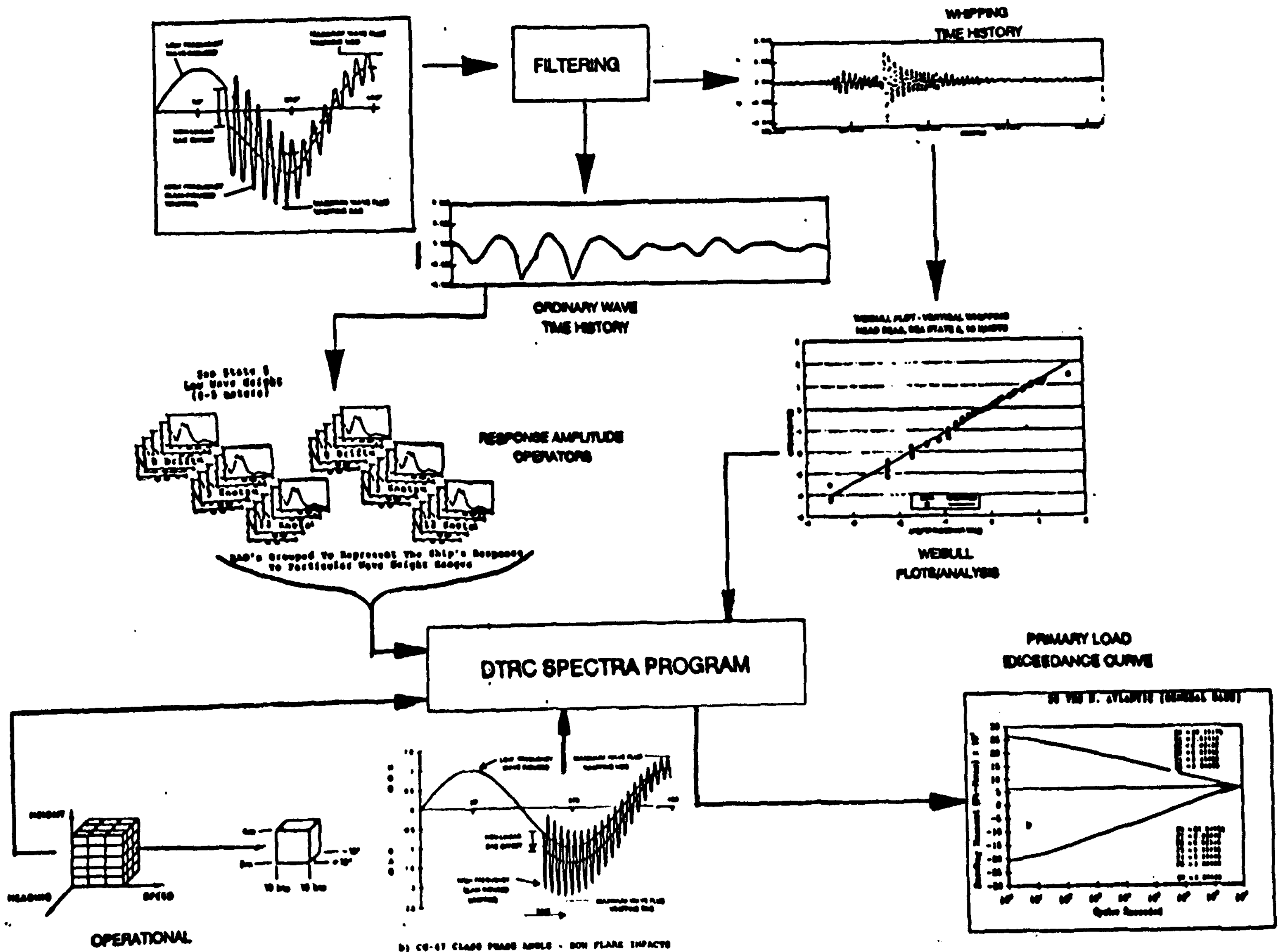
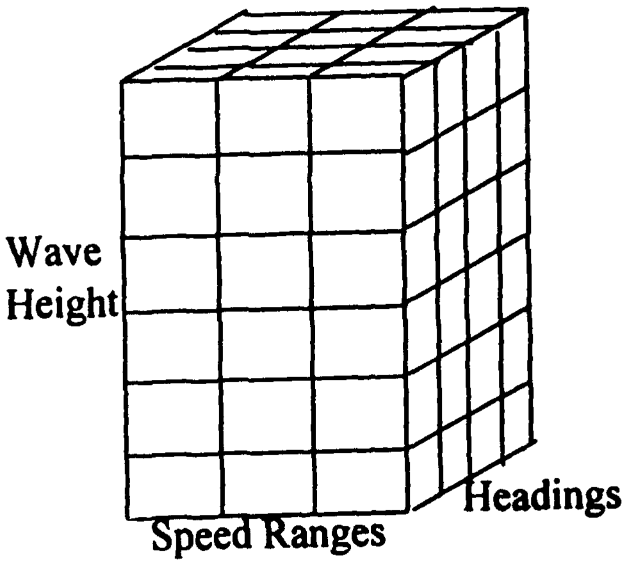


FIGURE 70 LIFETIME LOADS ANALYSIS METHODOLOGY

OPERATIONAL PROFILES



• WAVE HEIGHT	16
• HEADINGS	5
• SPEED RANGES	>3
• SPECT. SHAPES	11
• TOTAL	2640

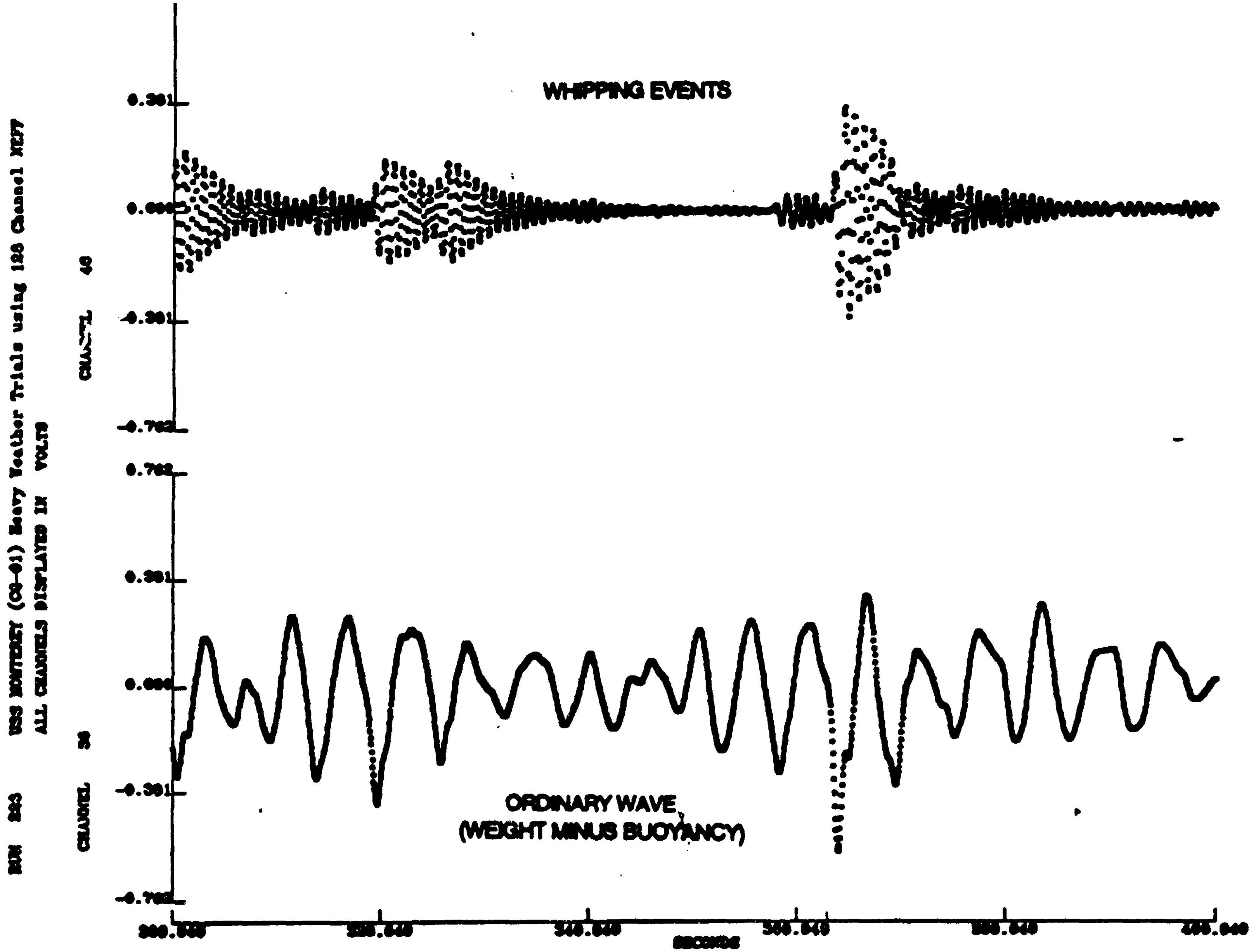
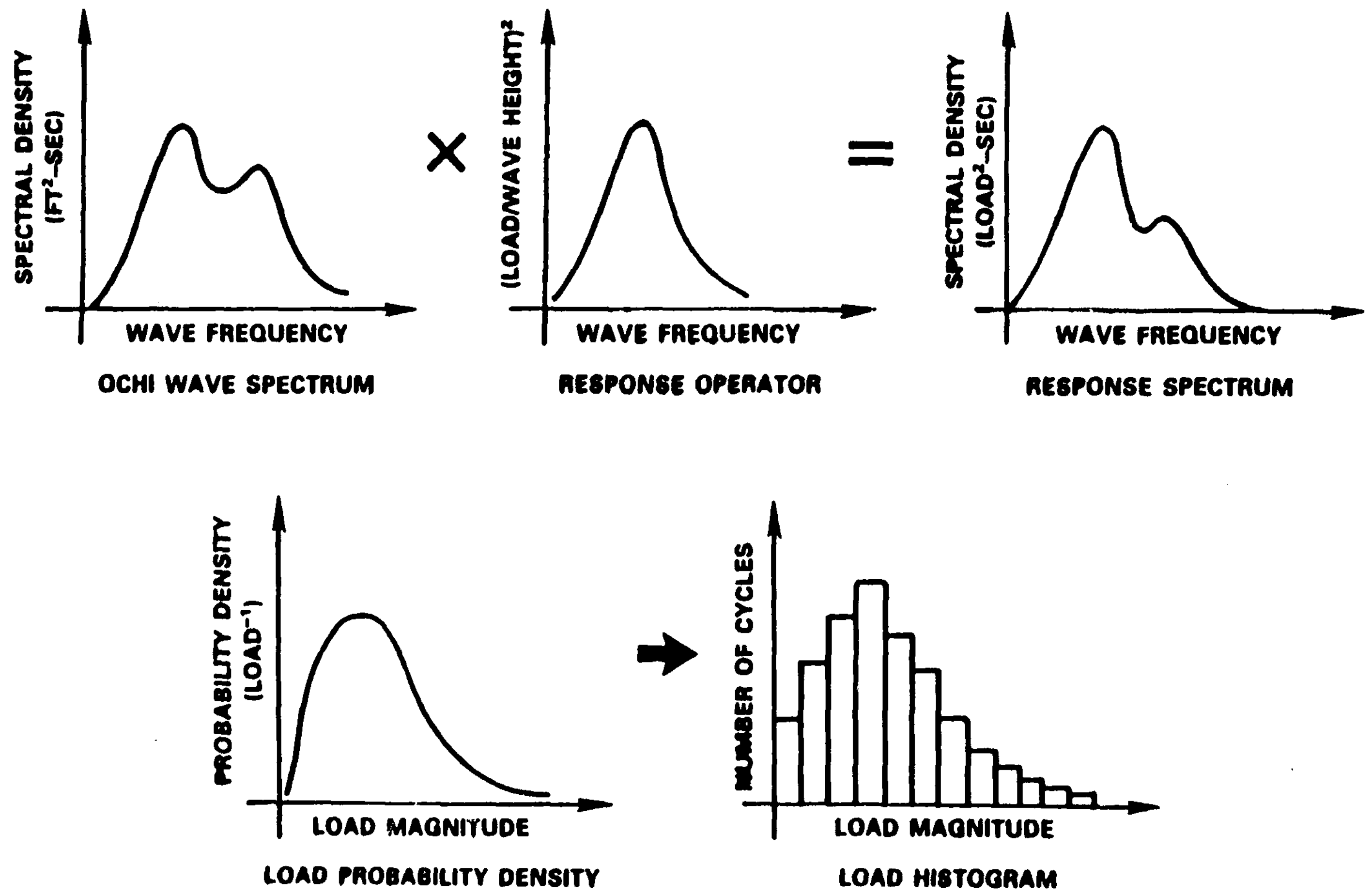


FIGURE 28 TYPICAL FILTERED TIME HISTORIES OF WAVE INDUCED BENDING, SHOWING ORDINARY WAVE AND WHIPPING

IV. CALCULATE SEAWAY BENDING MOMENTS AS FUNCTIONS OF TIME & SEA STATE



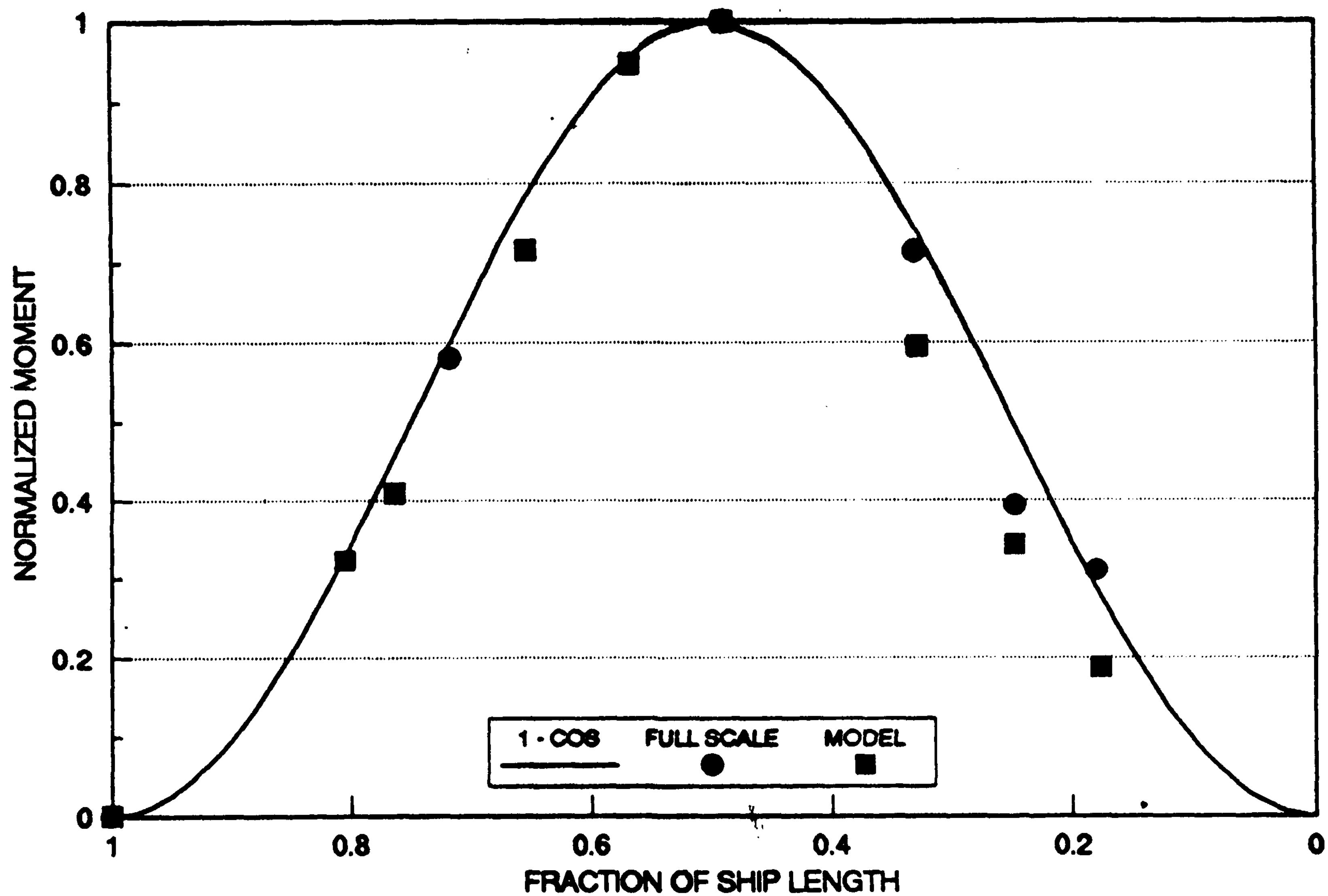


FIGURE 21 ORDINARY WAVE VERTICAL BENDING MOMENT DISTRIBUTION

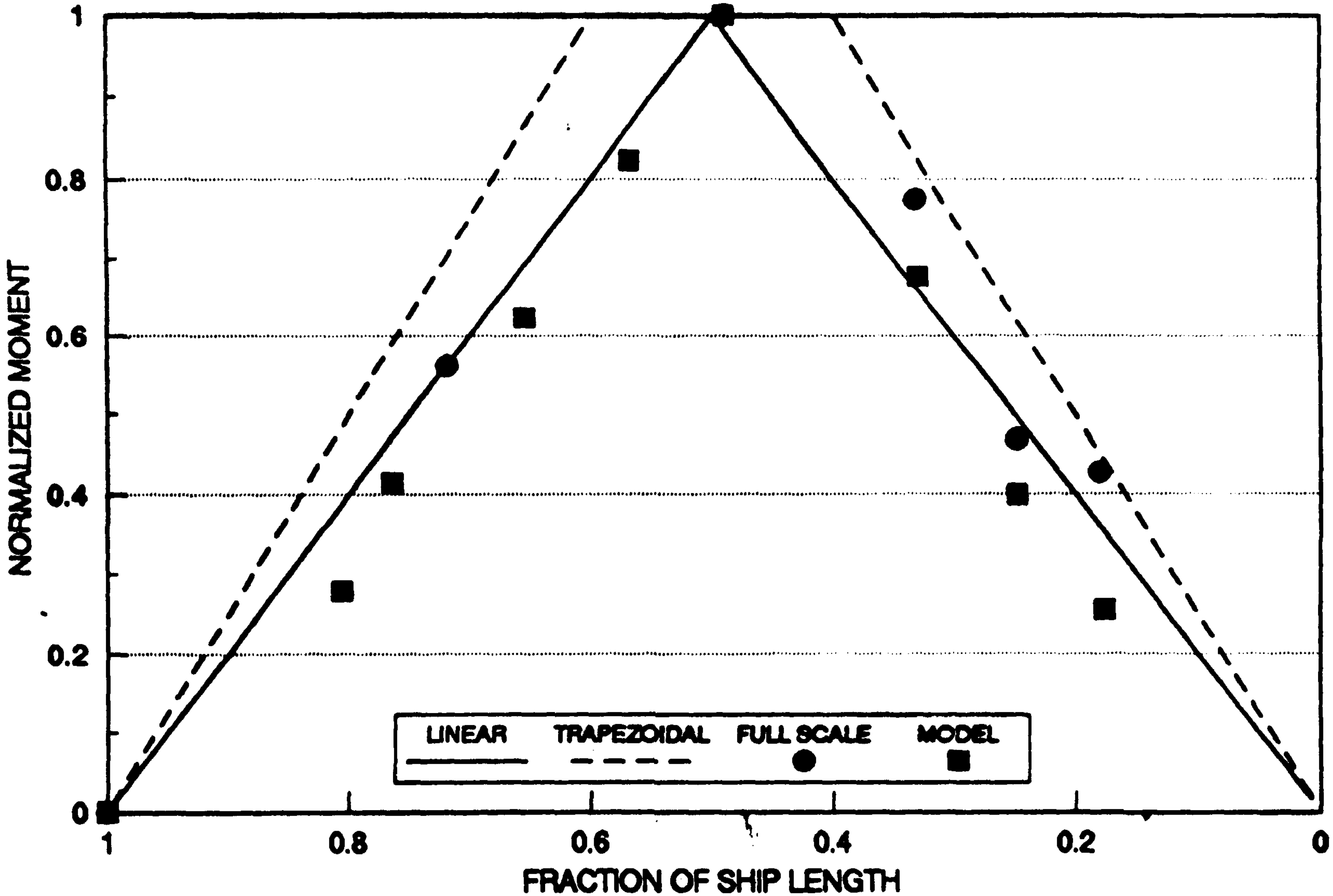


FIGURE 23 VERTICAL BENDING MOMENT DISTRIBUTION FOR WHIPPING ONLY

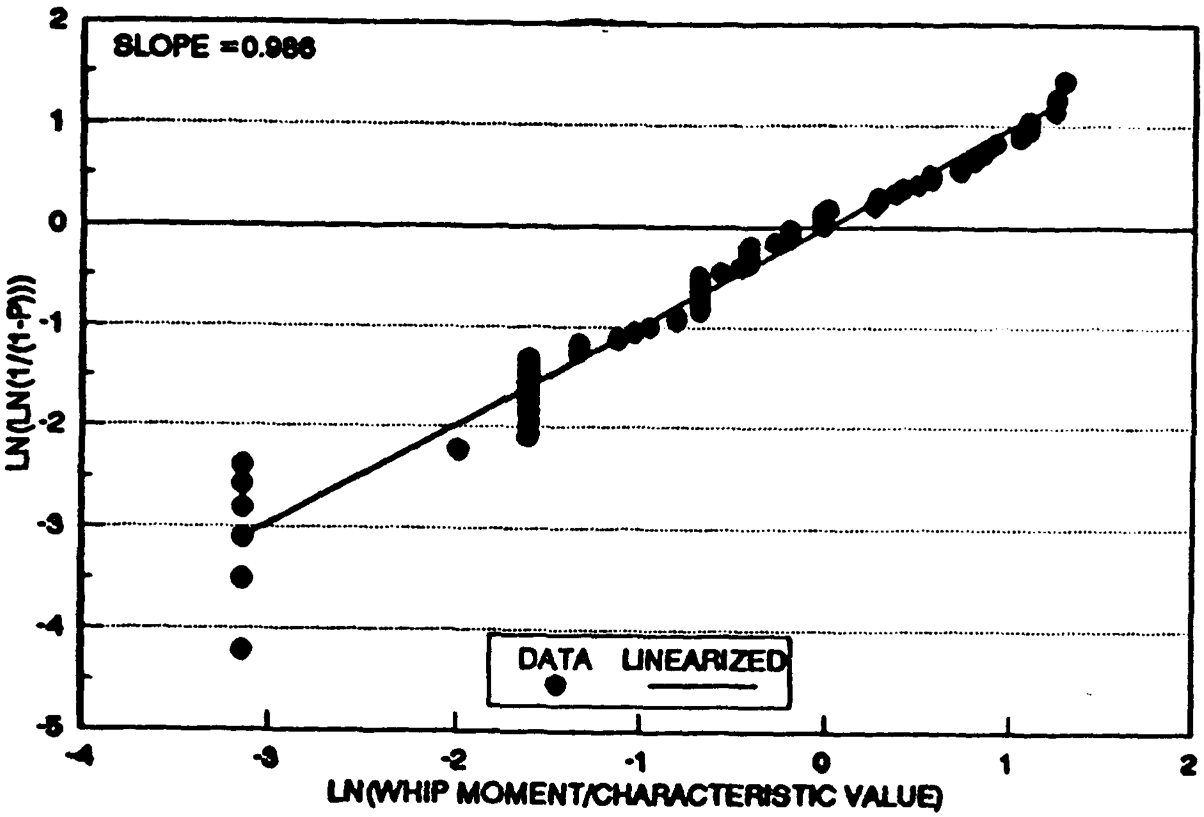


FIGURE 38 TYPICAL WEIBULL PLOTS OF CG-61 FULL SCALE MIDSHIP WHIPPING MOMENTS
c) VERTICAL BENDING, SEA STATE 6, 20 KNOTS, HEAD SEAS

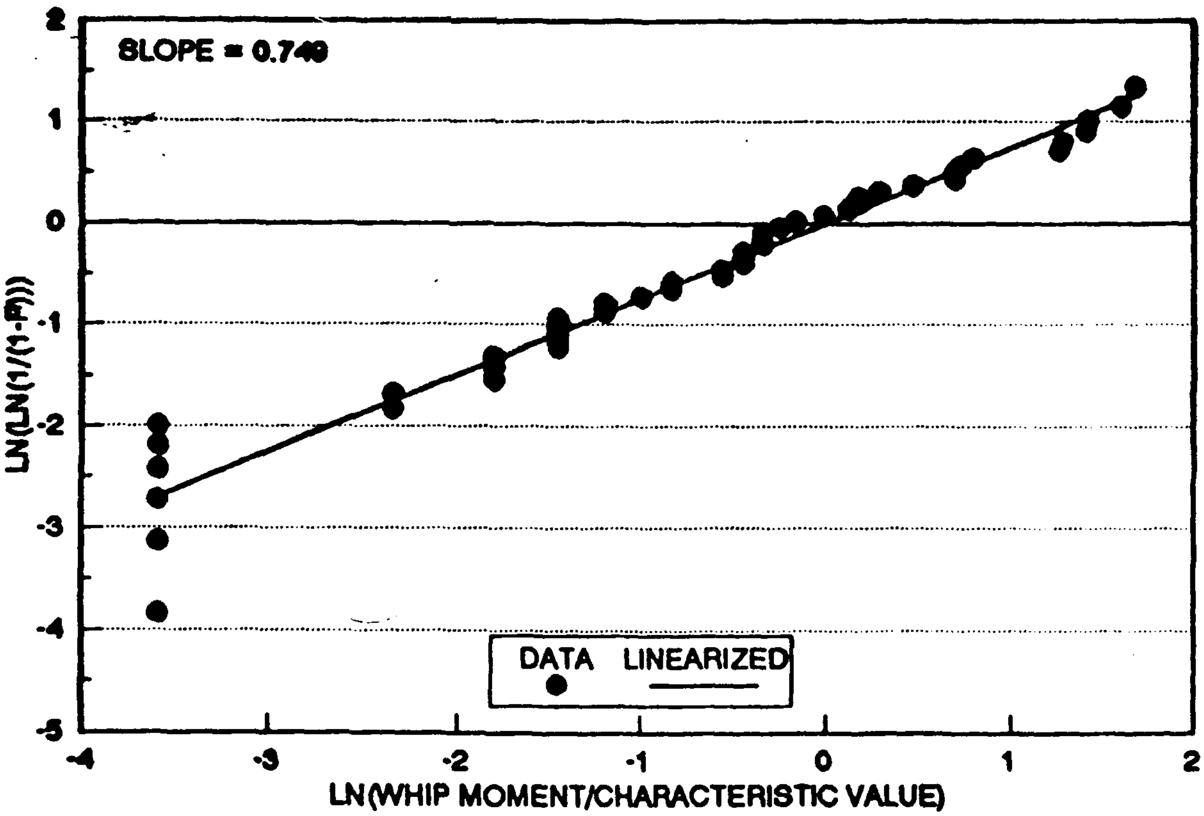


FIGURE 38 TYPICAL WEIBULL PLOTS OF CG-61 FULL SCALE MIDSHIP WHIPPING MOMENTS
d) VERTICAL BENDING, SEA STATE 6, 25 KNOTS, HEAD SEAS

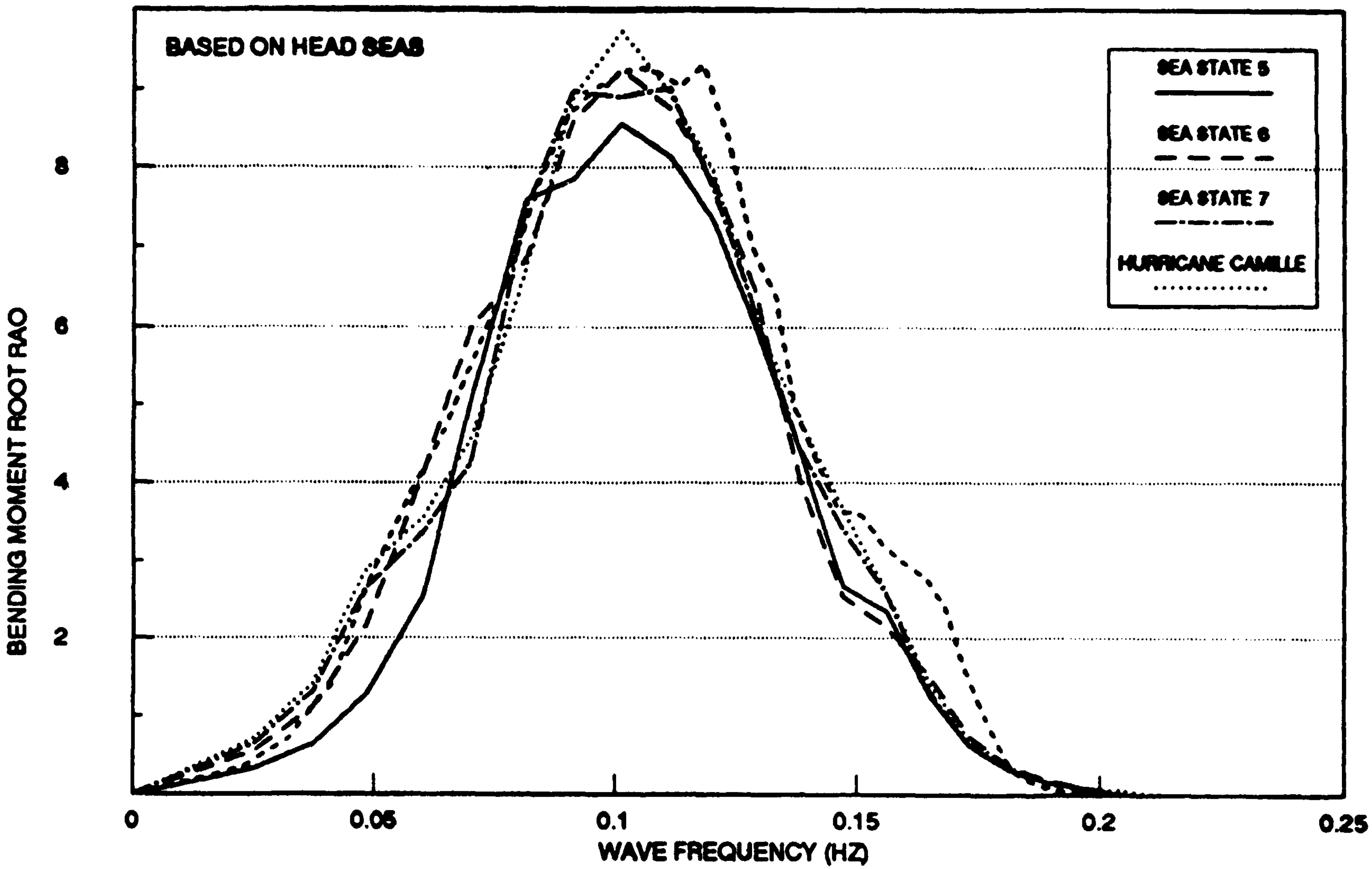
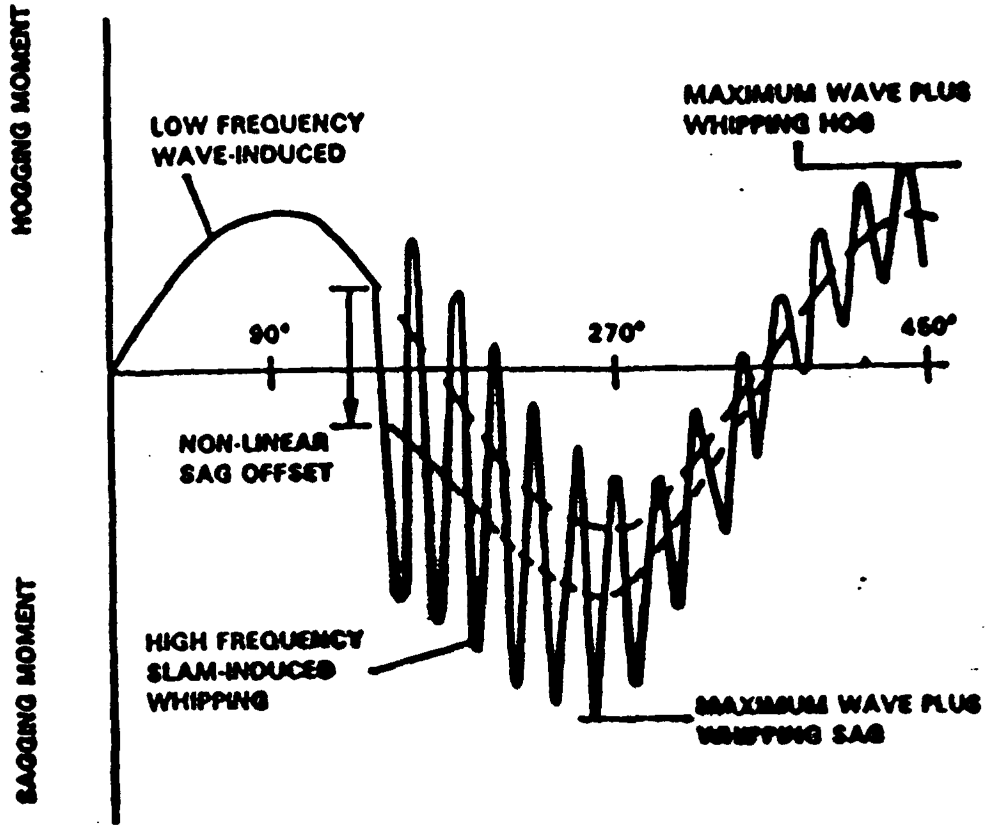
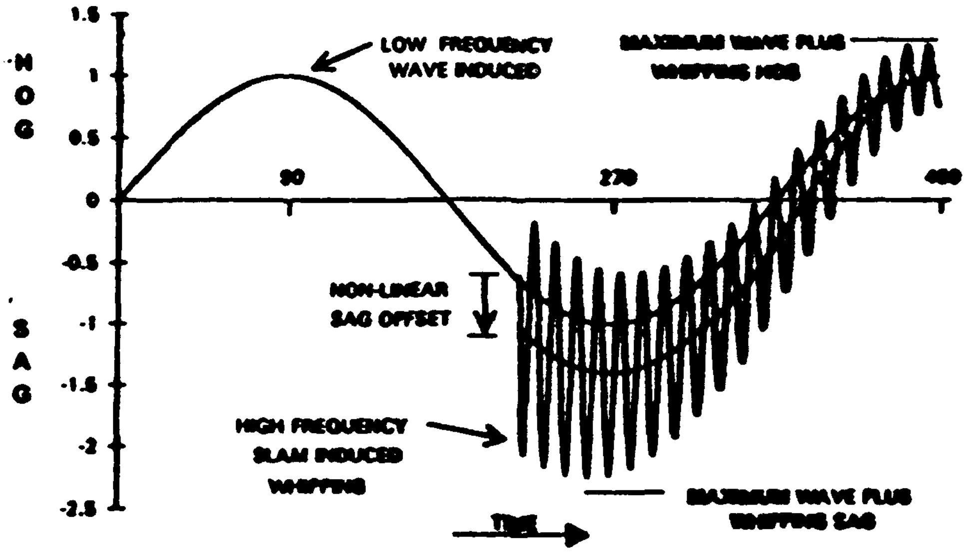


FIGURE 57 CG-47 CLASS MODEL BASED VERTICAL BENDING MOMENT ROOT RAO'S AS A FUNCTION OF SEA STATE



a) TRADITIONAL HULL BOTTOM WAVE IMPACT

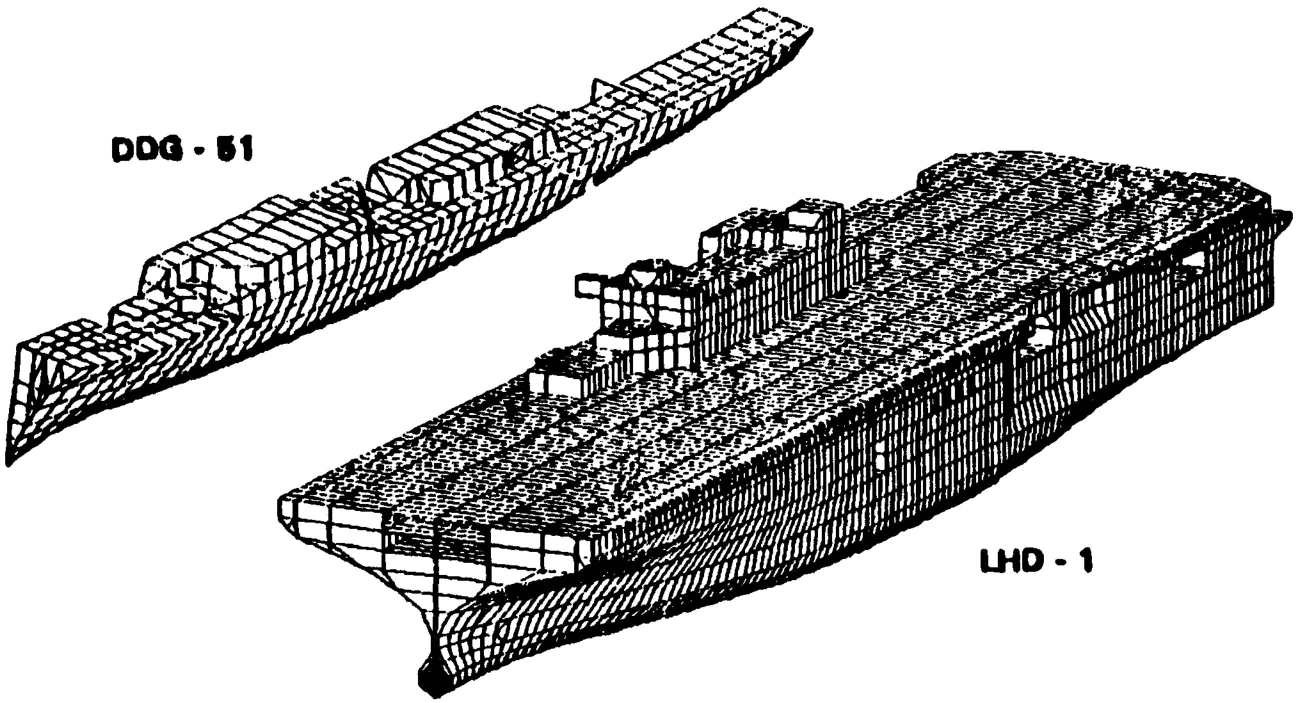


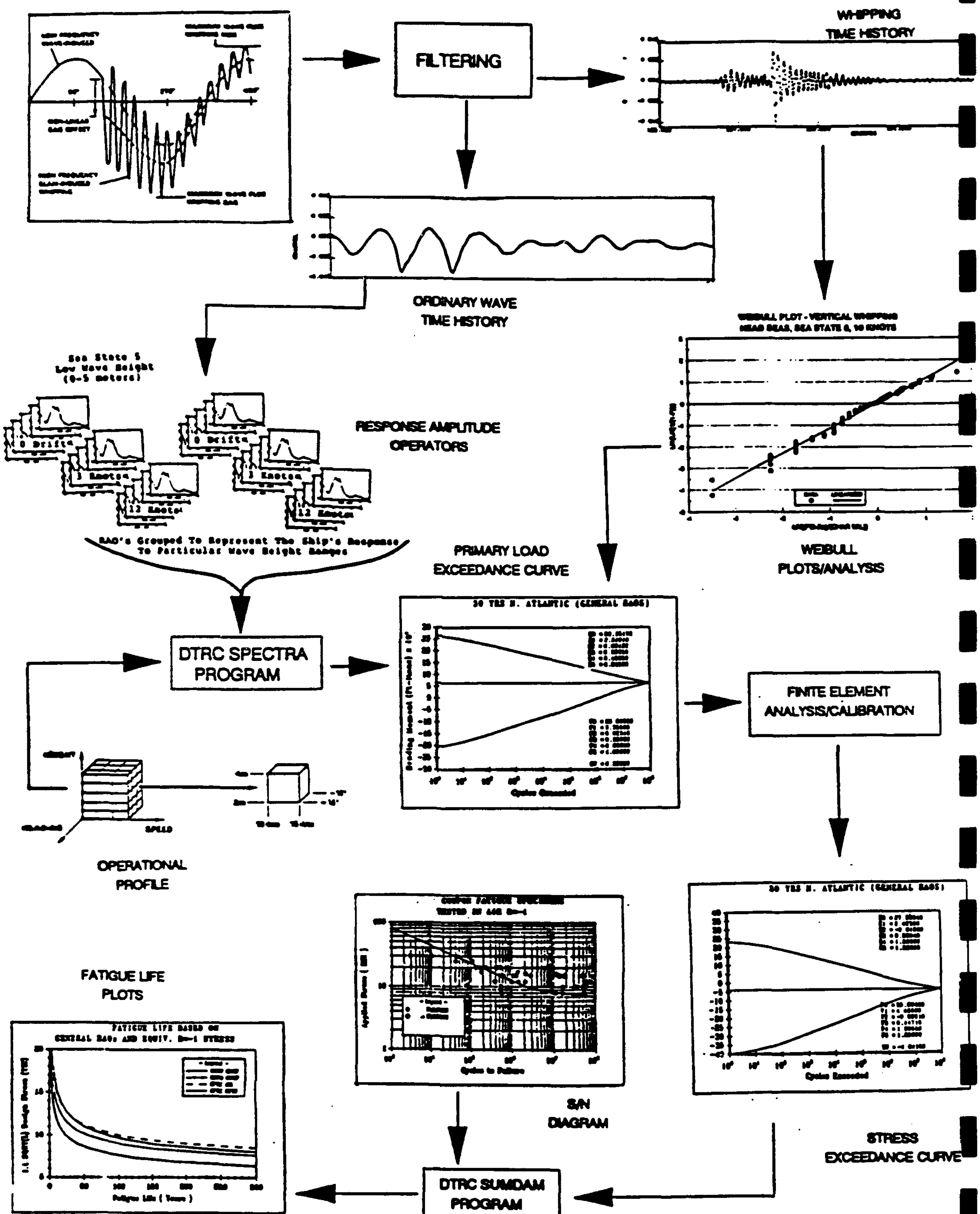
b) CG-47 CLASS PHASE ANGLE - BOW FLARE IMPACTS

FIGURE 27 TYPICAL BENDING MOMENT TIME HISTORIES SHOWING PHASE ANGLES BETWEEN ORDINARY WAVE AND WHIPPING

NSWC Naval Surface Warfare Center CARDEROCK DIVISION

Full Ship Models




FILTERING
WHIPPING
TIME HISTORY
ORDINARY WAVE
TIME HISTORY
Sea State 5
Low Wave Height
(0-5 meters)
0 Drifts
3 Knots
12 Knots
RESPONSE AMPLITUDE
OPERATORS
RAO's Grouped To Represent The Ship's Response
To Particular Wave Height Ranges
WEIBULL
PLOTS/ANALYSIS
PRIMARY LOAD
EXCEEDANCE CURVE
30 YRS N. ATLANTIC (GENERAL RAOS)
Cycles Exceeded
DTRC SPECTRA
PROGRAM
FINITE ELEMENT
ANALYSIS/CALIBRATION
OPERATIONAL
PROFILE
SPEED
FATIGUE LIFE
PLOTS
S/N
DIAGRAM
Cycles to Failure
STRESS
EXCEEDANCE CURVE
DTRC SUMDAM
PROGRAM
Fatigue Life (Years)

Virginia Tech

INTEGRATED SHIP STRUCTURAL DESIGN METHODOLOGY

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Program Objectives

- Leveraging of past work in ship structural design conducted in the US into a practical, computer-based system.
- Resolution of important technical requirements related to fatigue, ultimate strength and reliability which will facilitate the implementation of a reliability-based design system.
- Utilization of ongoing efforts in predicting ship motions and loads.
- Preserving the technical leadership of the US in ship structural design capabilities.
- Providing advances in the safety and cost effectiveness of military and commercial ship structures, offering dual-use of the integrated, computer-based structural design tool which will be produced by the research efforts.
- Enabling the rational, reliability-based structural design of ships from first principles, so that new or advanced geometries (e.g. double hulls, SWATHs) can be practically designed by non research groups such as shipyards and naval architects.

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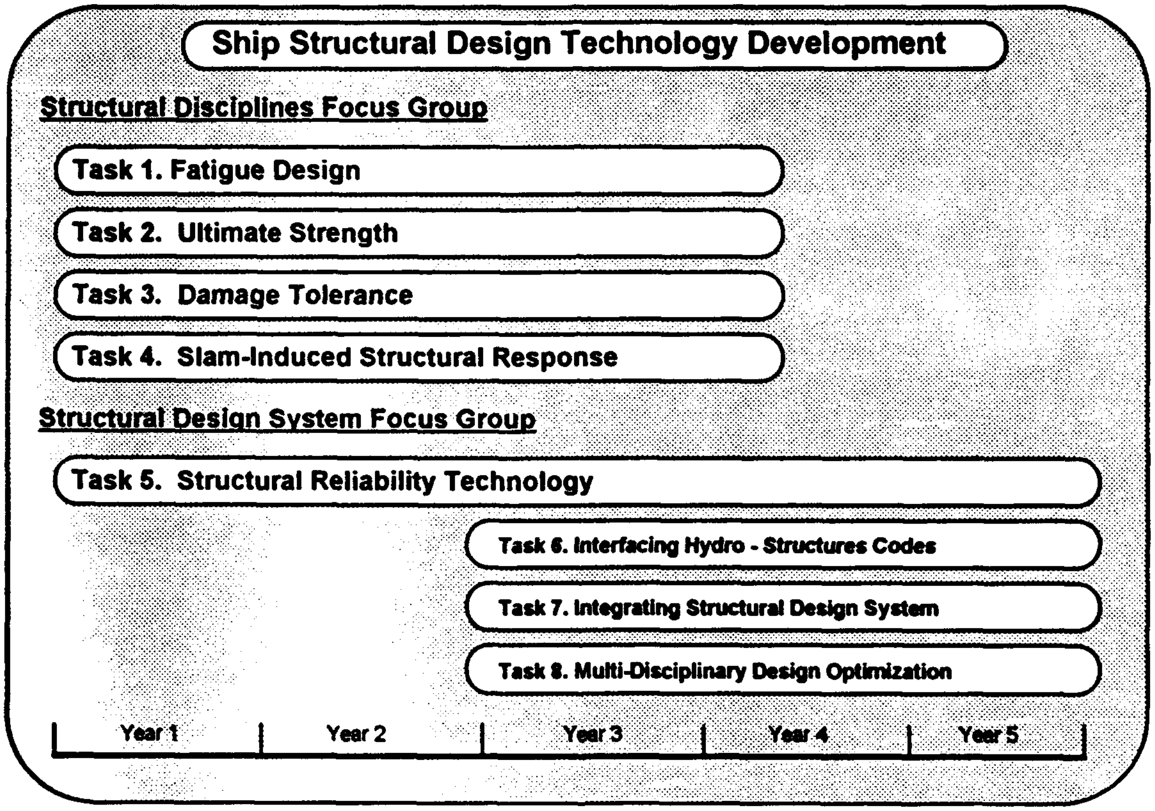
Overview of Research and Development Program



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Years One/Two Tasking

Task 1. Develop a Fatigue Design Procedure

Task 2. Improve Ultimate Strength Analysis Methods

Task 2. Develop Damage Tolerance Design Technology

Task 4. Computation of Slamming-Induced Structural

Task 5. Structural Reliability Technology

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Task 1. Develop a Fatigue Design Procedure

- Reformulate/extend the analysis method and the design method to include fatigue
- Define fatigue-related load data as needed from a Hydro program (e.g. the influence of whipping)
- Define a modular interface with such a program
- Extend the stress analysis to get cyclic stress transfer functions; initially for stiffeners and frames

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Task 2. Improve Ultimate Strength Analysis Methods

- Member Level - top priority need: a better model for flexural-torsional buckling of beams

- Overall Level - further improve the current analysis model to better account for post-buckling stiffness

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Task 3 Develop Damage Tolerance Design Technology

- Define relevant load conditions for damage tolerant design (e.g. grounded and/or flooded conditions)
- Develop detailed plan for incorporation of Damage Tolerance Approach into design practice

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Task 4 Computation of Slamming-Induced Structural Responses

- Define local and global slamming-related load data needed from Hydro program(s)
- Later phases will address the computation of structural response to slamming and the interfacing between slam load prediction codes and structural response codes

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Task 5 Structural Reliability Technology

- Define the basic technology requirements for a reliability-based format for ship structures
- Assess the state-of-art for the technology requirements and determine/prioritize the development needs
- Develop a plan for ensuing years' efforts
- Identify and establish liaison with current relevant research

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Implementation of Results

- Overall Objective: To incorporate these developments, together with the other components of design, into an integrated structural design system which can be used by the ship design community
- ONR has decided that *MAESTRO* will be the vehicle for implementing the results into an existing structural design tool
- *MAESTRO* integration with other design tools has been selected under the Maritech Modular Tanker Consortium project
- Our team will also be participating in the US-Norway research project *Dynamic Analysis of Surface Ships.*

ONR Workshop - July '98

An Integrated Calculation Package for Ship Hull Slamming & Wave-Induced Effects

Alaa Mansour

Naval Architecture & Offshore Engineering
University of California, Berkeley

A. Mansour

ONR Workshop - July '98

Objective

Develop an integrated package for ship hull extreme loads and stresses resulting from the COMBINED effects of slamming and wave-induced loads.

A. Mansour

Applications

- *Definition of extreme load effects for design with account of load phasing.*
- *Calculation of hull girder reliability.*
- *Forecasting & hindcasting combined slam and wave-induced response.*
- *Defining operability envelope*
- *Correlations against service data.*

New Features

The package will specifically account for:

- *Nonlinearities in the load effects.*
- *Phasing of slamming with respect to the wave-induced loads.*
- *Clustering effects (grouping).*

ONR Workshop - July '94

[illegible]lopment of the [illegible] methodology

- Seaway described by narrow-banded spectrum.
- Motions kinematics linearly related to wave (given by an RAO).
- Dynamic transients are small and evolve slowly - structure responds to local wave sinusoid with no effects of previous wave.

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Task 1: (Cont.)

- Wave spectrum is replaced by time domain wave components.
- Time domain calculations require only deterministic calculations of response to regular waves of given amplitude and frequency.
- Joint density of wave amplitude and frequency leading to slamming determined by Slepian regression.

A. Mansour

1: (Cont.)

Seaway component:

$$\eta(t;a,\omega) = a \cos(\omega t + \varepsilon)$$

For any choice of a and ω process is stationary, ε uniform over $[0,2\pi]$.

Calculated responses due to wave should include both low-frequency wave-induced and high-frequency slamming-induced components.

Task 1: (Cont.)

The maximum response at a location is $M(a,\omega)$ for any choice of ε.

The response statistics are formed by weighing the $M(a,\omega)$ by the joint probability density $f(a,\omega)$ of various doublets of (a,ω).

(cont.)

The response statistics (n^{th} order moments) can be expressed:

$$E[M^n] = \int_{all\,a}\int_{all\,\omega} [M(a,w)]^n f(a,\omega)\, da\, d\omega$$

An approximation to the distribution function $F_M(m)$ can be obtained from a Hermite transformation of the first four moments.

Task 2. Determination of Slam Impact force

2 Slam Phenomena:

- Bottom impact - an initial, short-duration, high pressure transient ($\approx$ 0.01 sec).

  Ochi & Motter or Stavovy & Chuang give maximum bottom pressure, $p = K V^2$. (for Ochi & Motter, K is also a function of deadrise angle, β).

(Cont.)

- *Momentum transfer slamming - a longer-duration, low pressure transient (≈ 1 sec).*
  - *Leibowitz gives the bottom force as a function of time*

$$\frac{D}{Dt}\left[m(t)\frac{Dw_{rel}(t)}{Dt}\right]$$

where m(t) is the sectional added mass per unit length.

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Task 3: Calculation of the Structural Response

- *Primary loadings will be considered (beam response).*
- *Both shear and bending stresses will be calculated.*
- *Both horizontal and vertical bending will be considered.*

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Dynamic equations for motion of the beam (including shear deformations):

$$\frac{\partial}{\partial x}\left[EI(x)\left(1 \ldots \frac{\partial}{\partial t} \right) \ldots \right] \ldots \frac{\partial}{\partial t}\right)\left(\frac{\partial w}{\partial x} - \phi\right)$$

$$= m_s(x)\,\kappa^2 \frac{\partial^2 \phi}{\partial t^2}$$

$$\ldots \left[\ldots \frac{\partial}{\partial t}\right)\left(\frac{\partial w}{\partial x} - \phi\right)\right] = m_s(x)\frac{\partial^2 w}{\partial t^2} - F(x,t)$$

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Task ... (cont.)

The solution may be approximated as a finite sums:

$$\ldots \sum_{i=0} \ldots(t)\,\gamma_i(x)$$

where α_i and γ_i are ortho-normalized eigenfunctions.

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- *Slamming process not exactly a Poisson Process.*
- *Slamming impacts tend to occur in clusters and are NOT independent.*
- *Both horizontal and vertical bending will be considered.*

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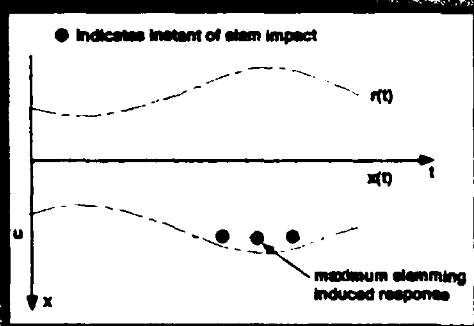

Narrow-banded Gaussian process with associated envelope

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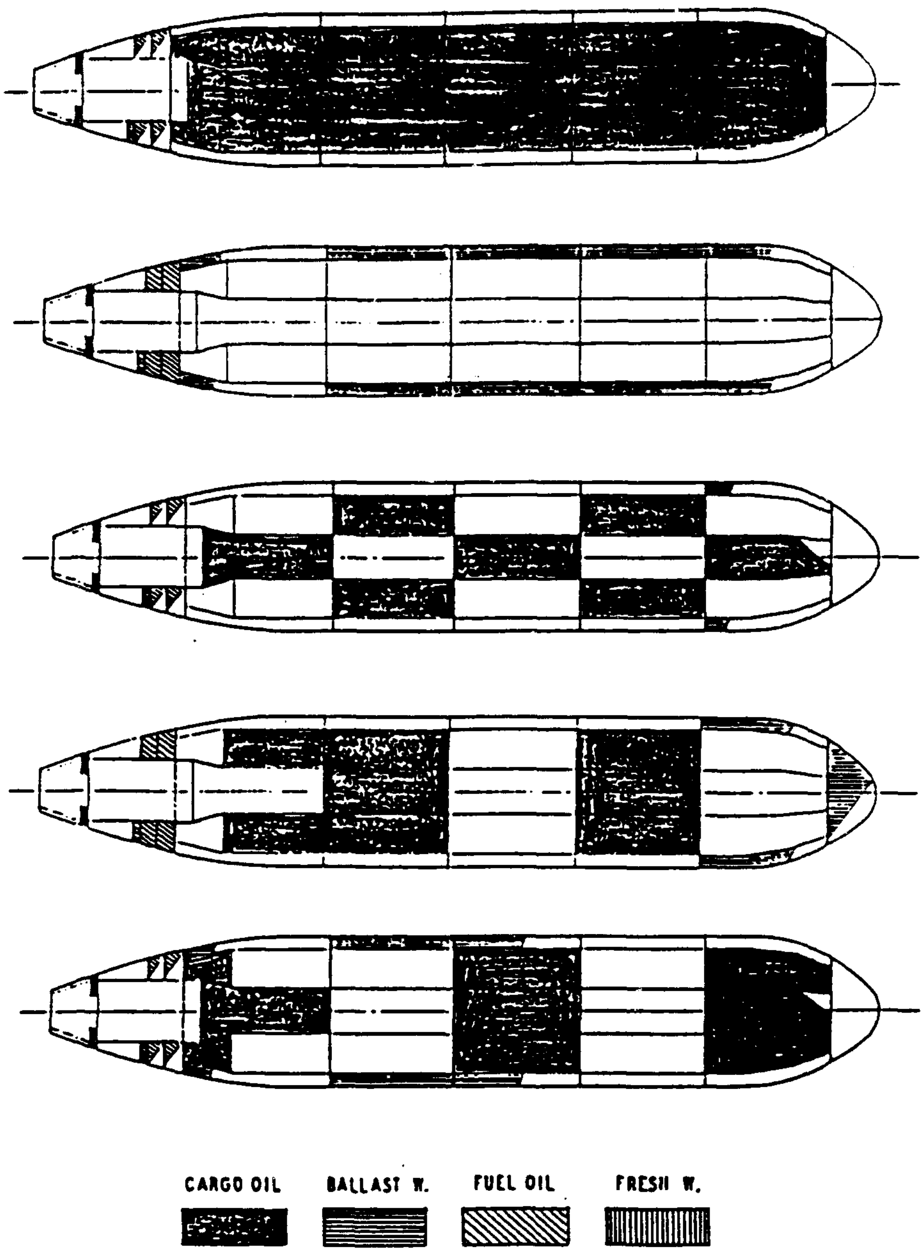
Dominant Load Parameters

- Vertical Bending Moment
- Vertical Acceleration
- Lateral Acceleration
- Roll Motion

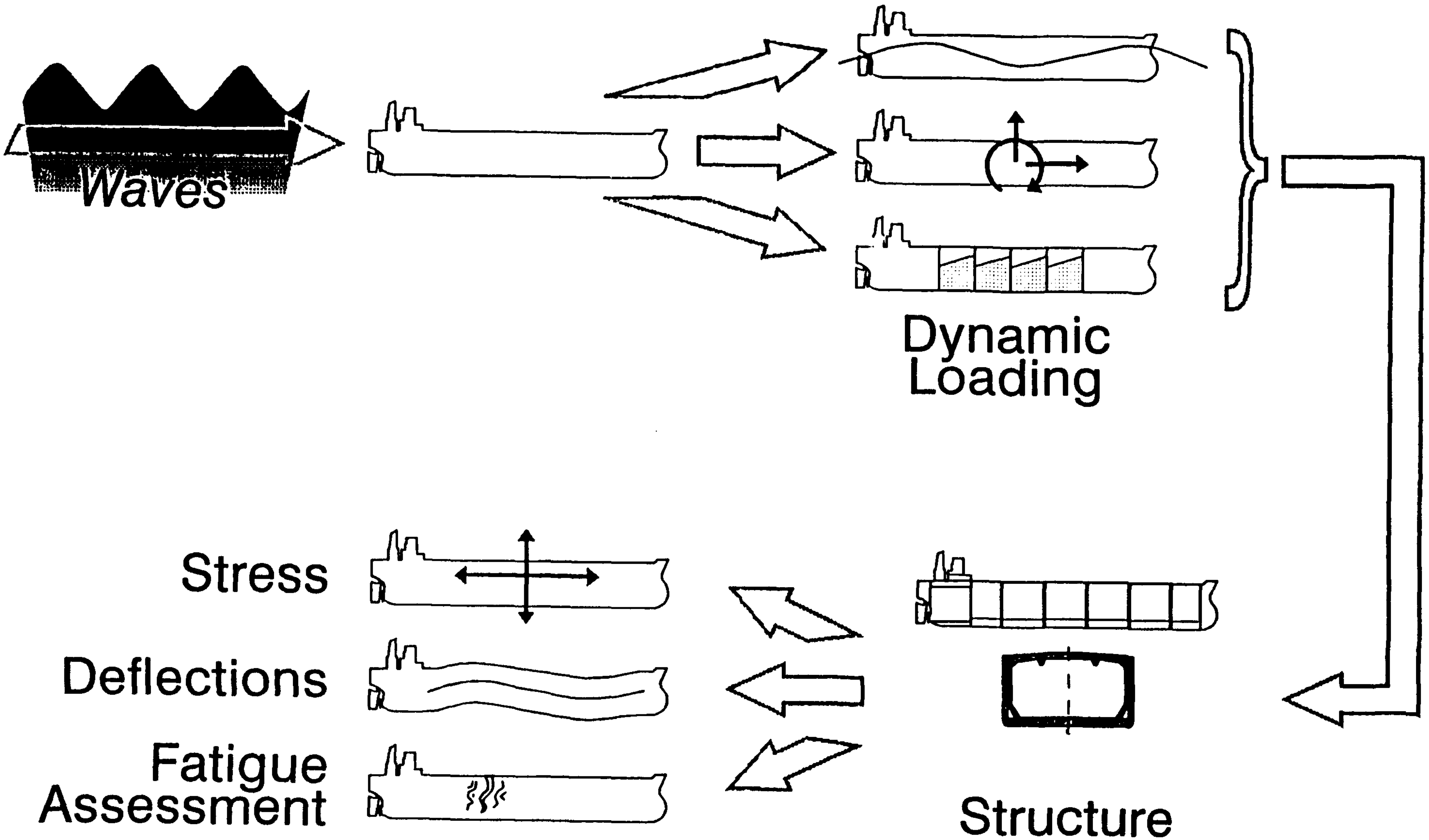
Representative Cargo Loading Conditions



Structural Load Cases

- Cargo loading conditions
- Dominant load parameters
- Equivalent wave systems
- Structural members of interest

Dynamic Loading Approach



Introduction of DLA

For Added Margin of Safety

- The design is based on an analysis with explicit dynamic loads.
  - Realistic shipmotions in waves
  - Extreme but realistic dynamic loads

- Scantling CAN NOT be less than Steel Vessel Rules requirements.
  - Increase in scantlings where needed

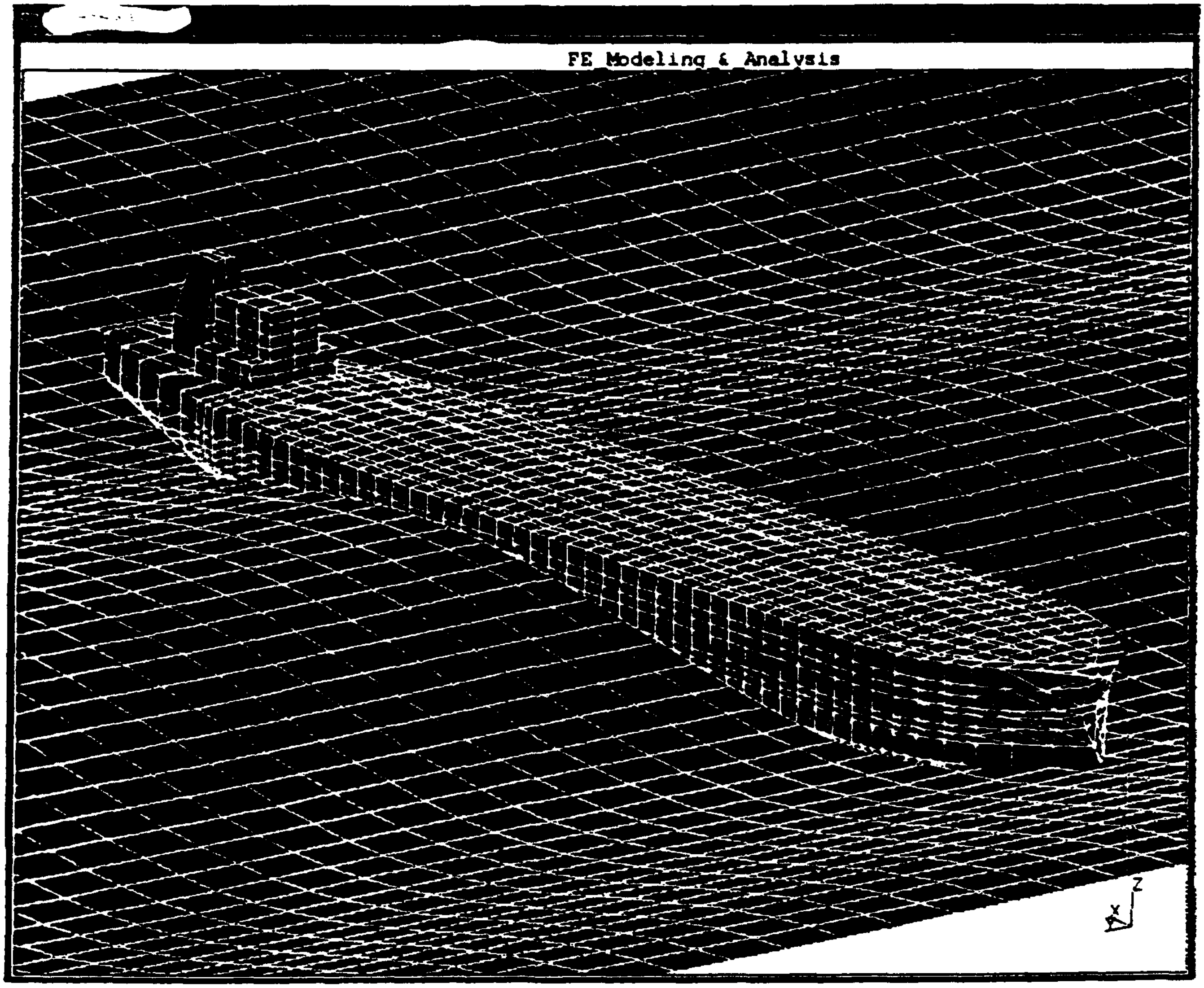
Current Practice

- Dynamic load is approximated by static load plus a factor.
- Factor is derived from service experience.
- The dynamic characteristics of individual vessel design are not considered.
- This results in generalized loading on vessel structure.

Background

- Classification Rules
- Direct Engineering Analysis
- Dynamic Loading Approach (DLA)

What are
the dynamic loads
acting upon vessels
at sea?



Equivalent Wave in Partial Loading (67% LOAD(C))

DLP : Maximum Hogging Condition

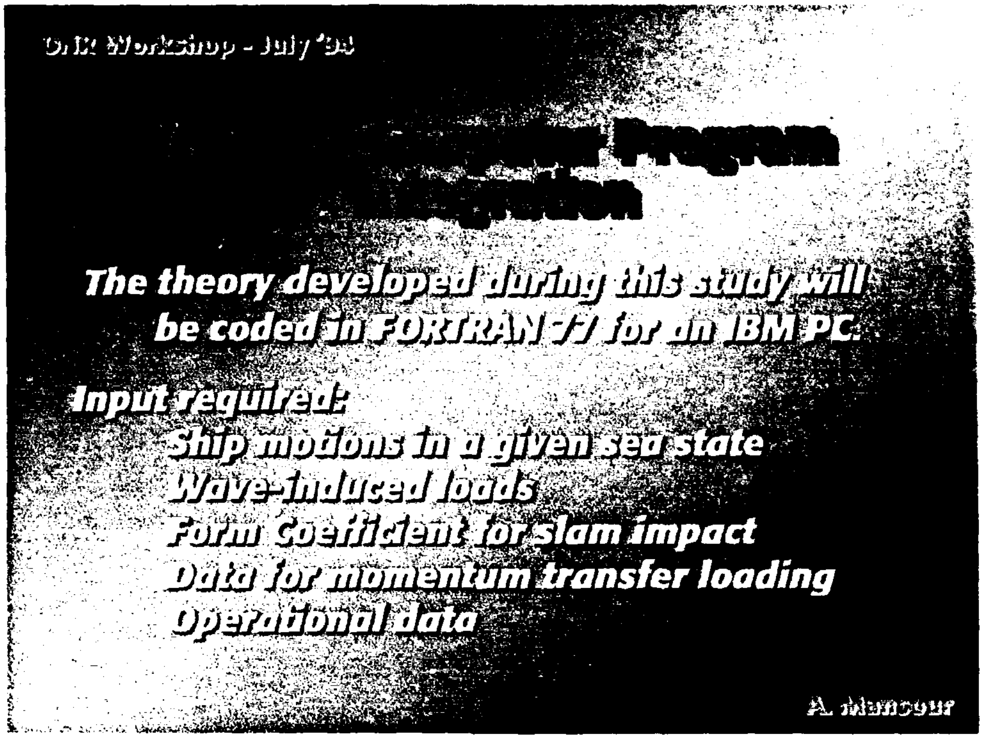
THE DYNAMIC LOADING APPROACH (DLA) FOR ANALYZING SHIP STRUCTURE

Presented at ONR Workshop on
Nonlinear Sea Loads and Ship Response
Ann Arbor, Michigan

7-8 July 1994

Yung S. Shin

AMERICAN BUREAU OF SHIPPING


Workshop - July
Program
The theory developed during this study will be coded in FORTRAN 77 for an IBM PC.
Input required:
Ship motions in a given sea state
Wave-induced loads
Form Coefficient for slam impact
Data for momentum transfer loading
Operational data
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Extreme Value Distribution of Response

The distribution of ... T is:

$F_{max\,M}(m) = P[$ ... $P[\max_{T\backslash T_{slam}} M \leq m]$

... $\geq m]$

... $P[\max_{T\backslash T_{slam}} M \geq m]$

... condition, $T/T_{slam} = T - T_{slam}$

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Task 5. (Cont.)

The extreme value distribution of maximum slamming and wave-induced response during the time period T is given as a Poisson pulse process for the maximum slam in each envelopes:

... $(m)]\,\nu T]$

Where: ν is the mean rate of "qualified" envelope excursions of level u (i.e., envelopes which have at least one slam).

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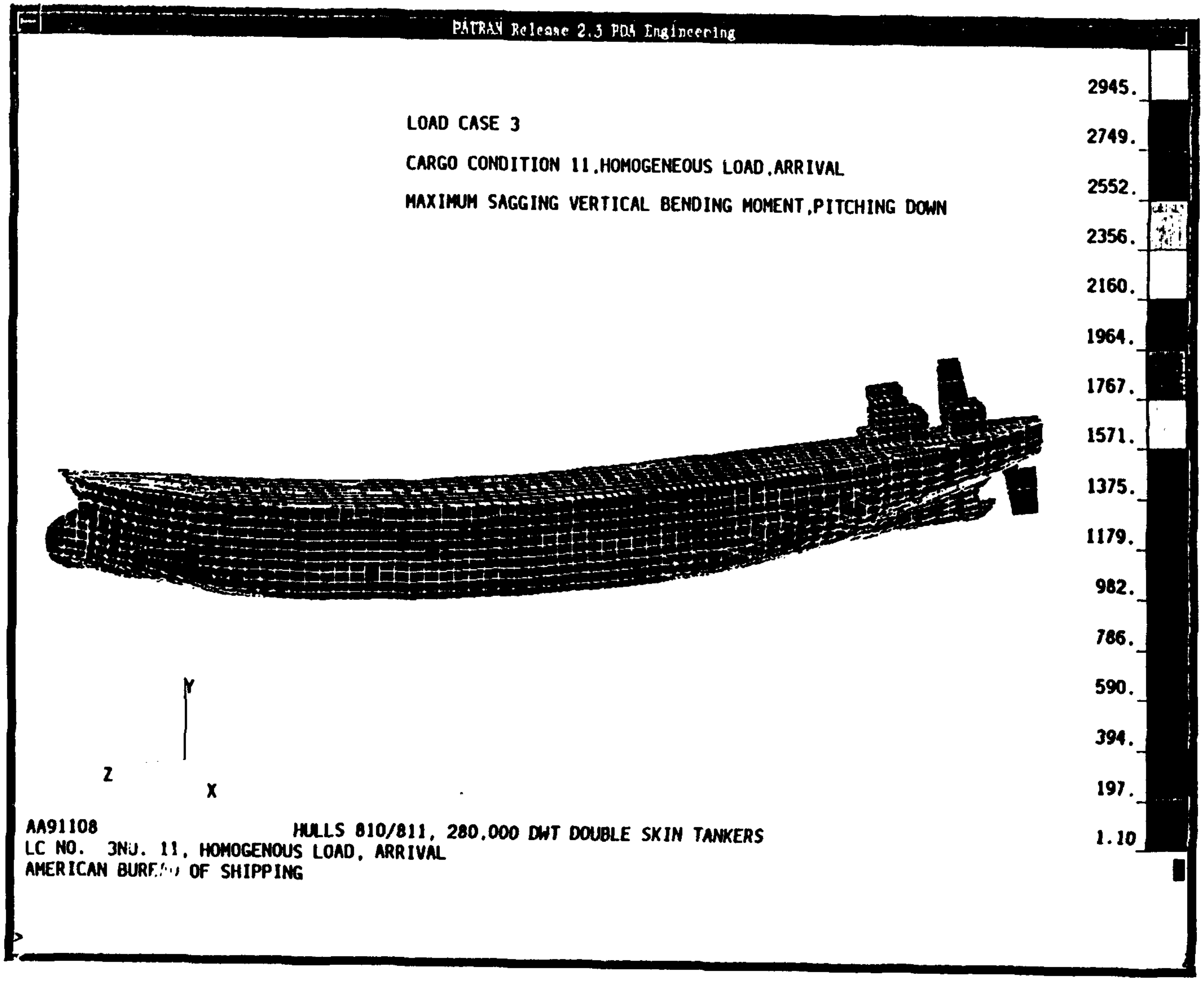
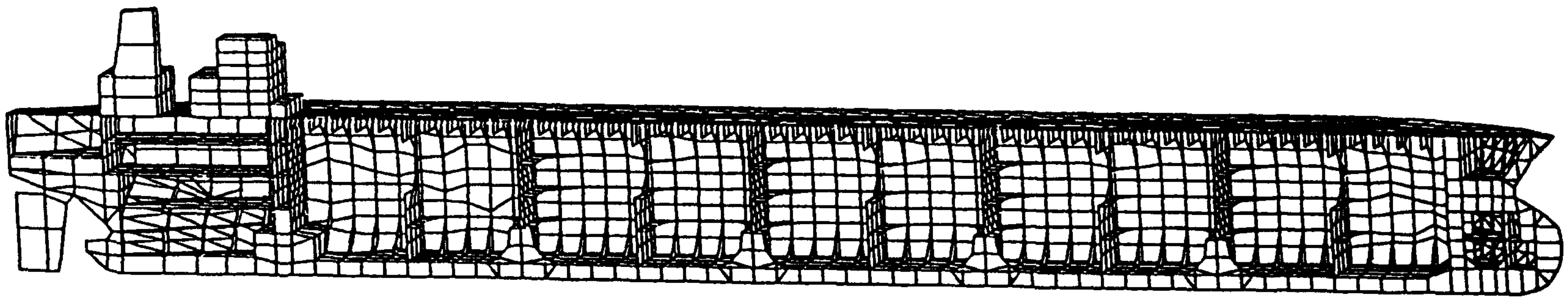
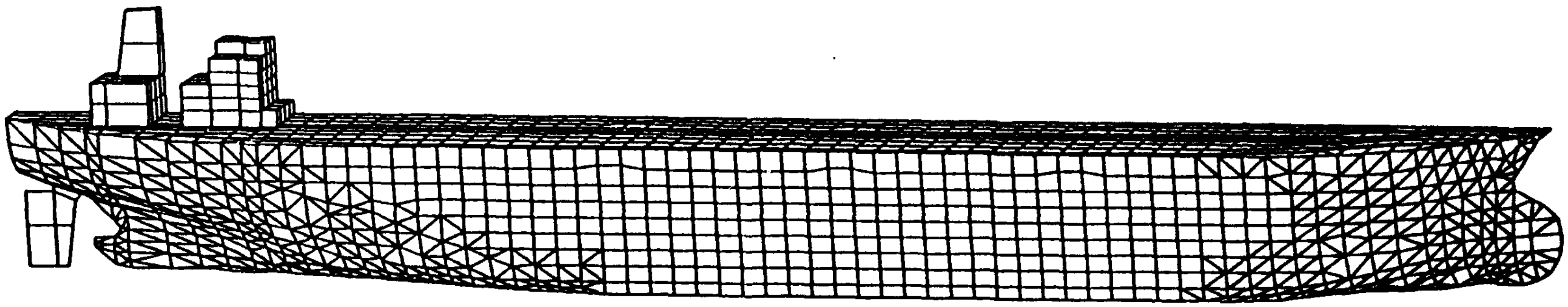
Typical 3-D Global Analysis Result
Max. sagging Condition



A Typical 3-D FEM Model of a Tanker





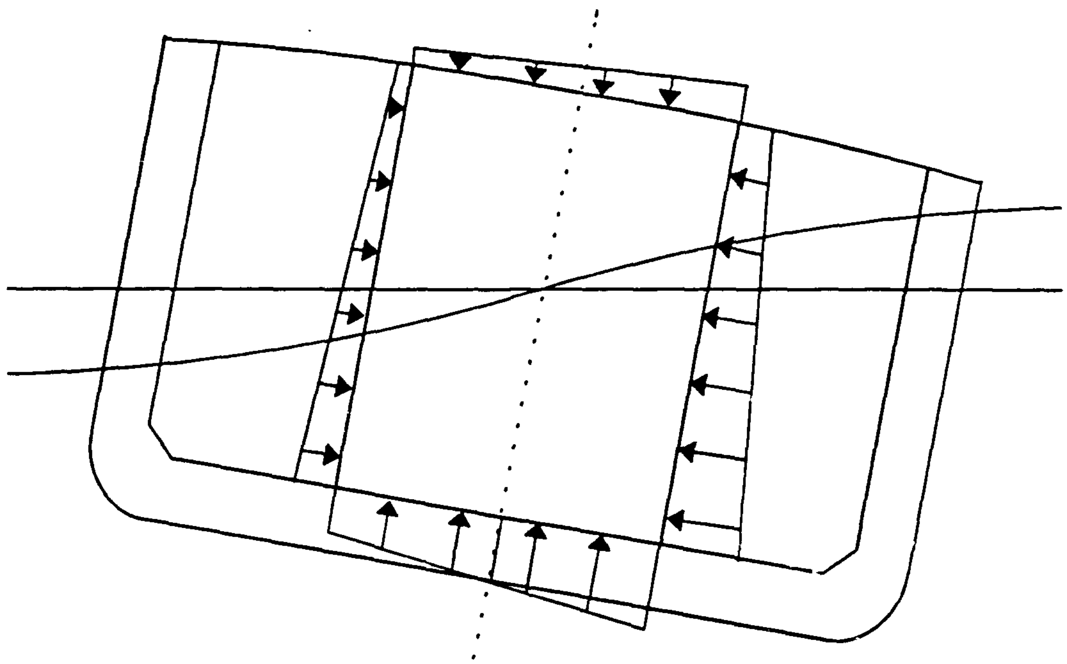
3-D Global FEM Analysis

- Extent of Finite Element Model
- Types of Finite Elements
- All Load Components to be Applied
- Decomposition of Loads

Loading for FEM Model

- Equilibrium Check
- Pressure Interpolation to FEM Model
- Boundary Force and Moment
- Fatigue Analysis

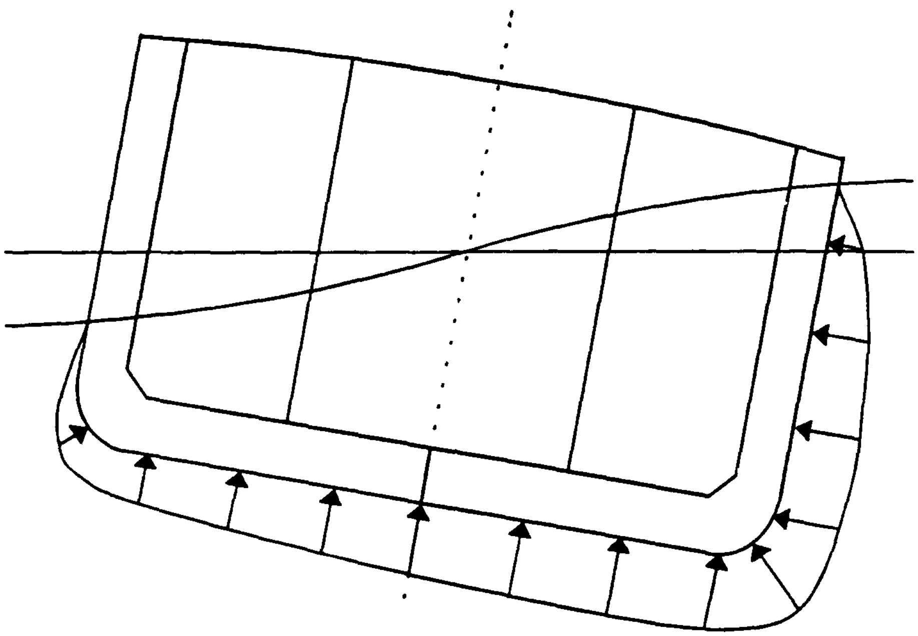
Internal Tank Pressure Distribution



Pressure Components due to

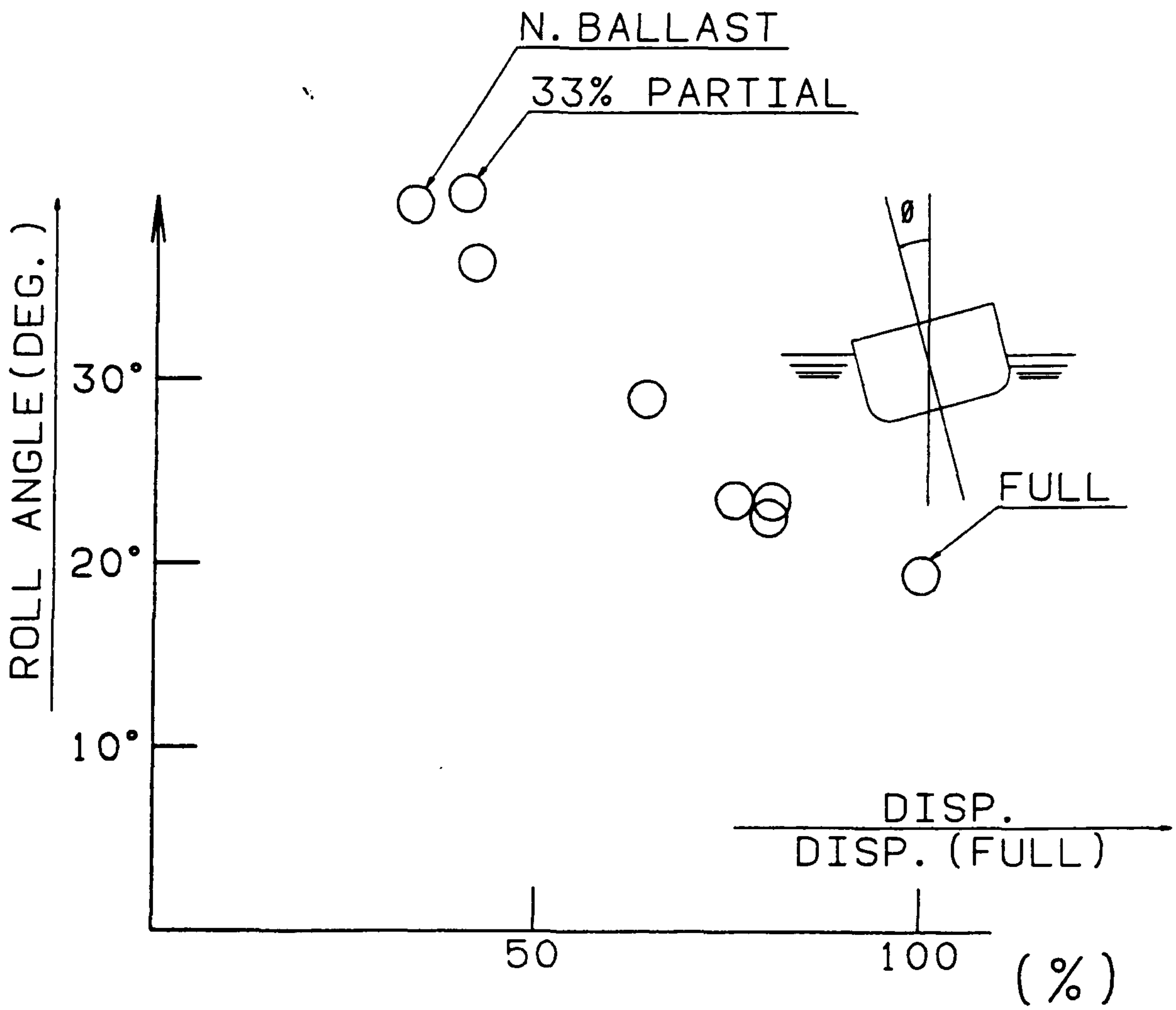
- Vapor Pressure
- Roll and Pitch Inclination
- Accelerations

External Pressure and Distribution



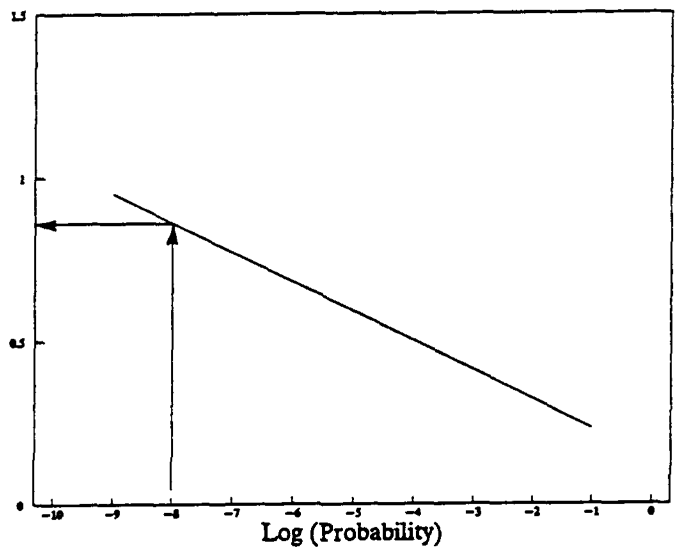
Pressure Components due to

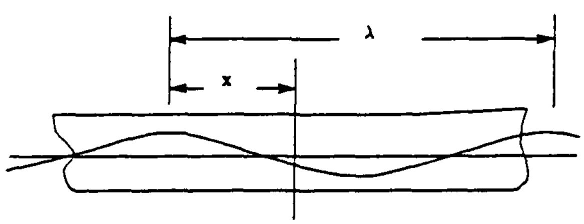
- Wave
- Vertical Motion
- Lateral Motion
- Roll



ROLL ANGLE V.S. SHIP'S DISPLACEMENT
(AT MAX. ROLL CONDITION)

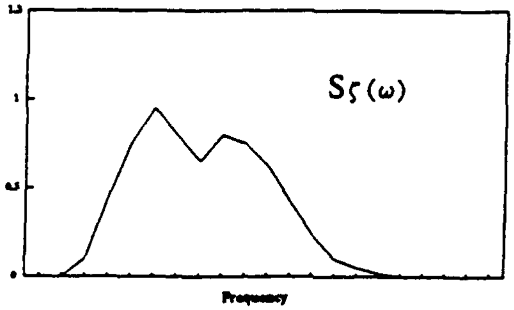
Long Term Response and Equivalent Wave System



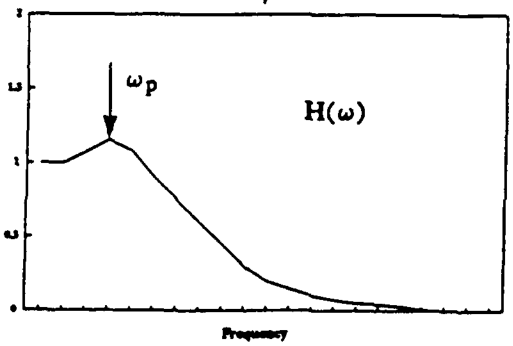


$$x = \frac{\lambda \epsilon}{-360 \cos \beta}$$

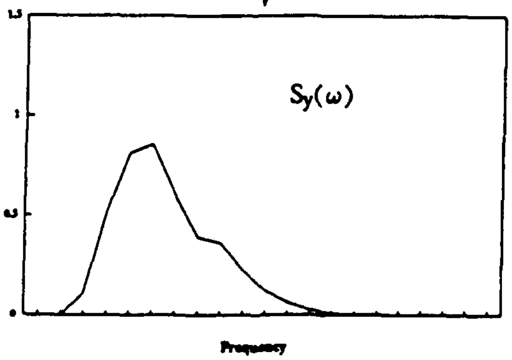
Short Term Response



WAVE SPECTRUM



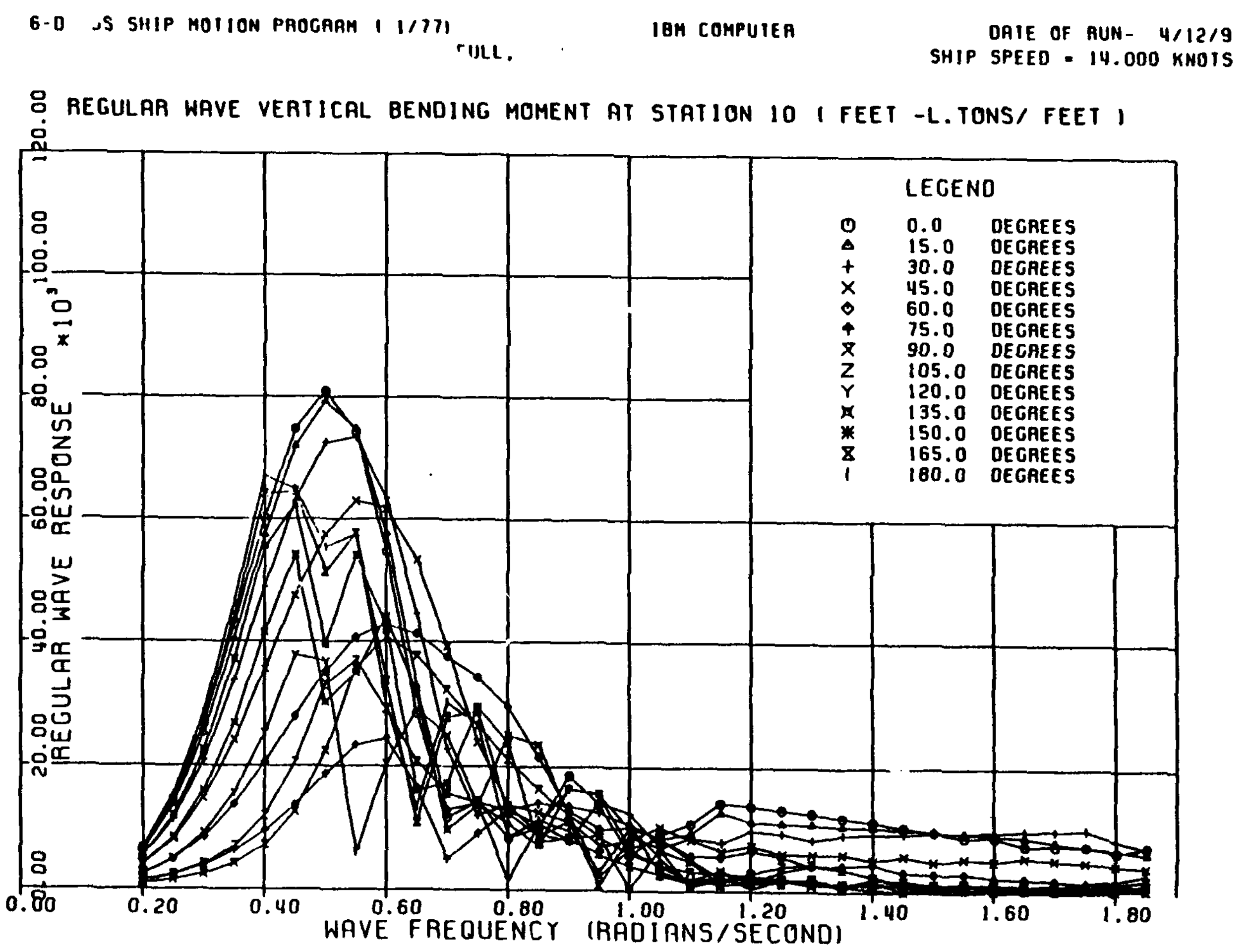
FRF OF DESIGN LOAD PARAMETER



RESPONSE SPECTRUM OF DLP

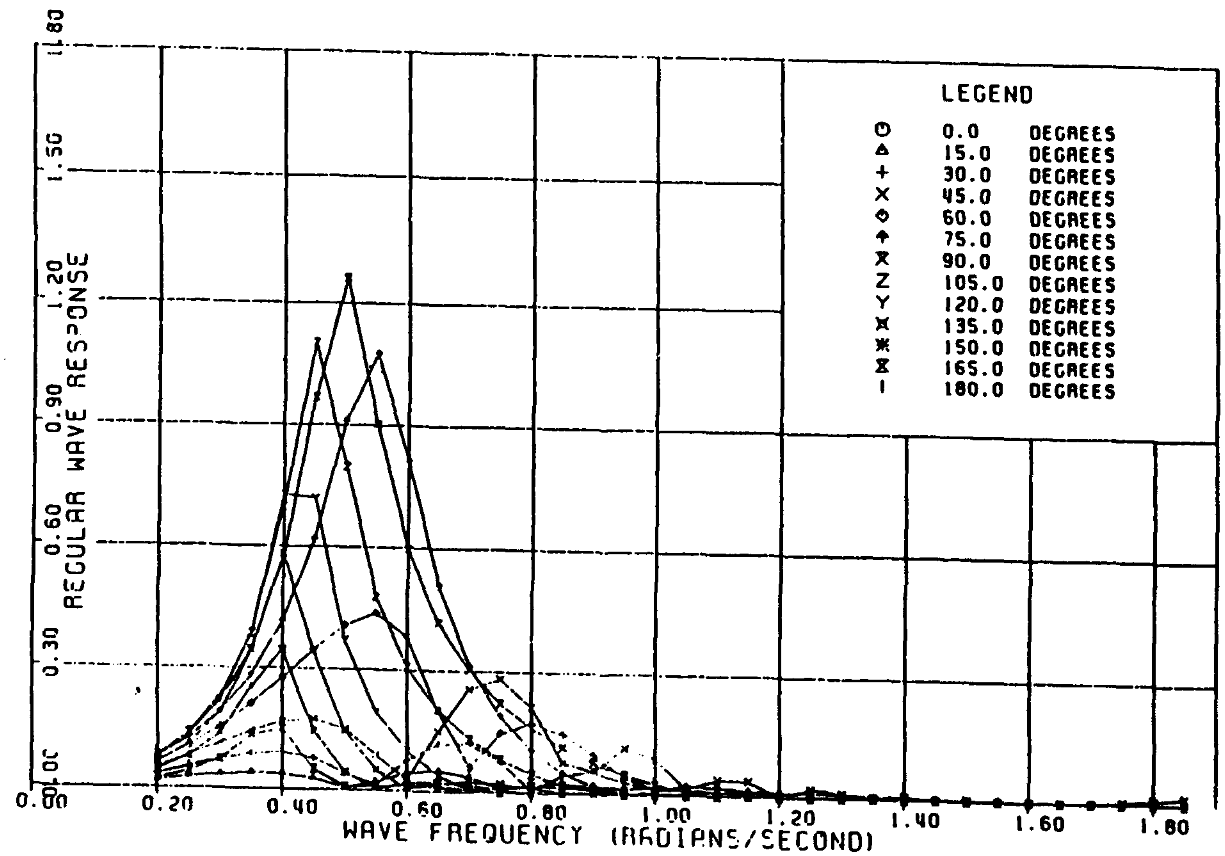
Representative Frequency Response Function

Vertical Bending Moment



Representative Frequency Response Function

Roll Motion

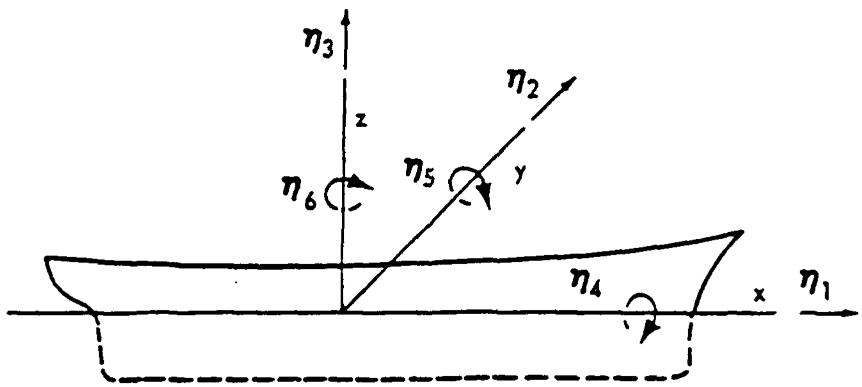


Equations of Motion

6-Degrees of Freedom Motion

- Surge
- Sway
- Heave
- Pitch
- Roll
- Yaw

$$\sum_{k=1}^{6} \left[(M_{jk} + A_{jk})\ddot{\eta}_k + B_{jk}\dot{\eta}_k + C_{jk}\eta_k\right] = F_j e^{i\omega t};\ j = 1\ldots 6$$



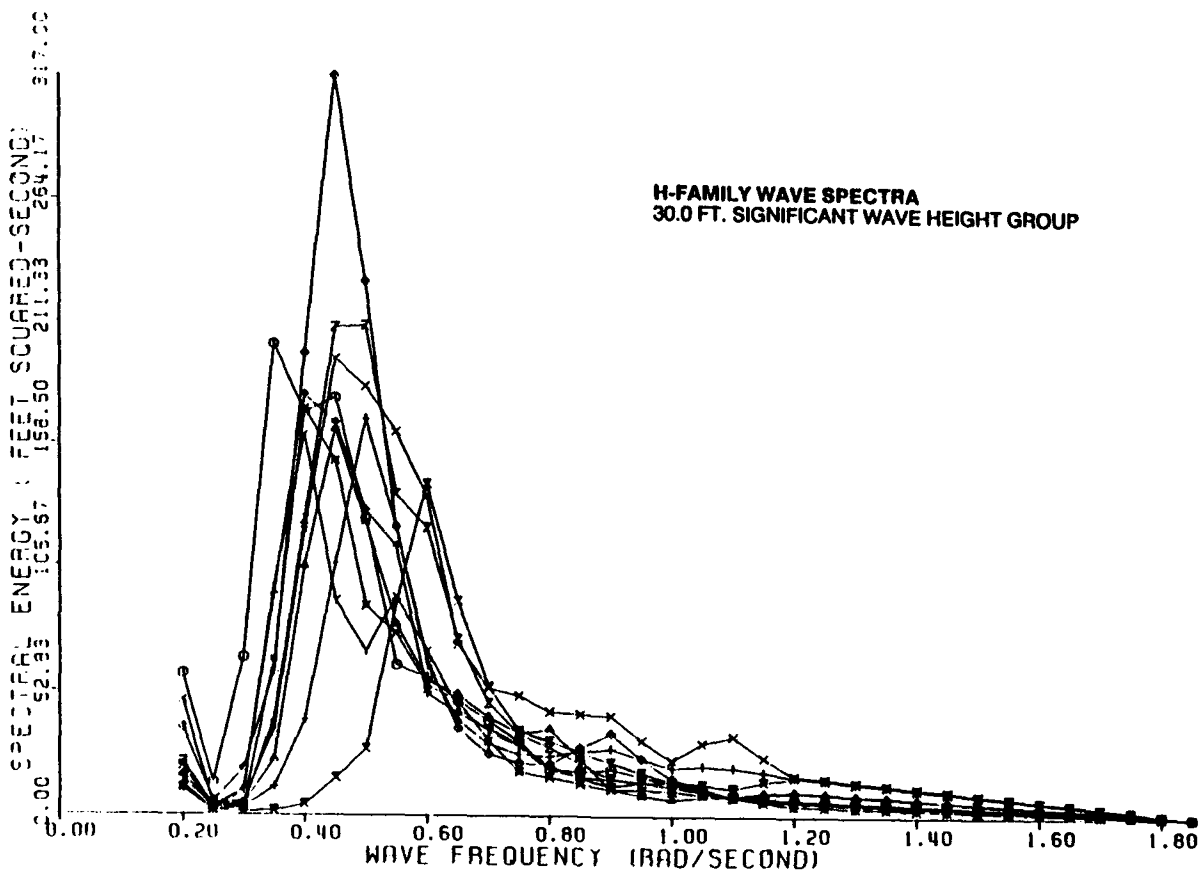
η_1 = surge η_3 = heave η_5 = pitch

η_2 = sway η_4 = roll η_6 = yaw

Representative Wave Spectra

H-Family

30 feet Significant Height Group



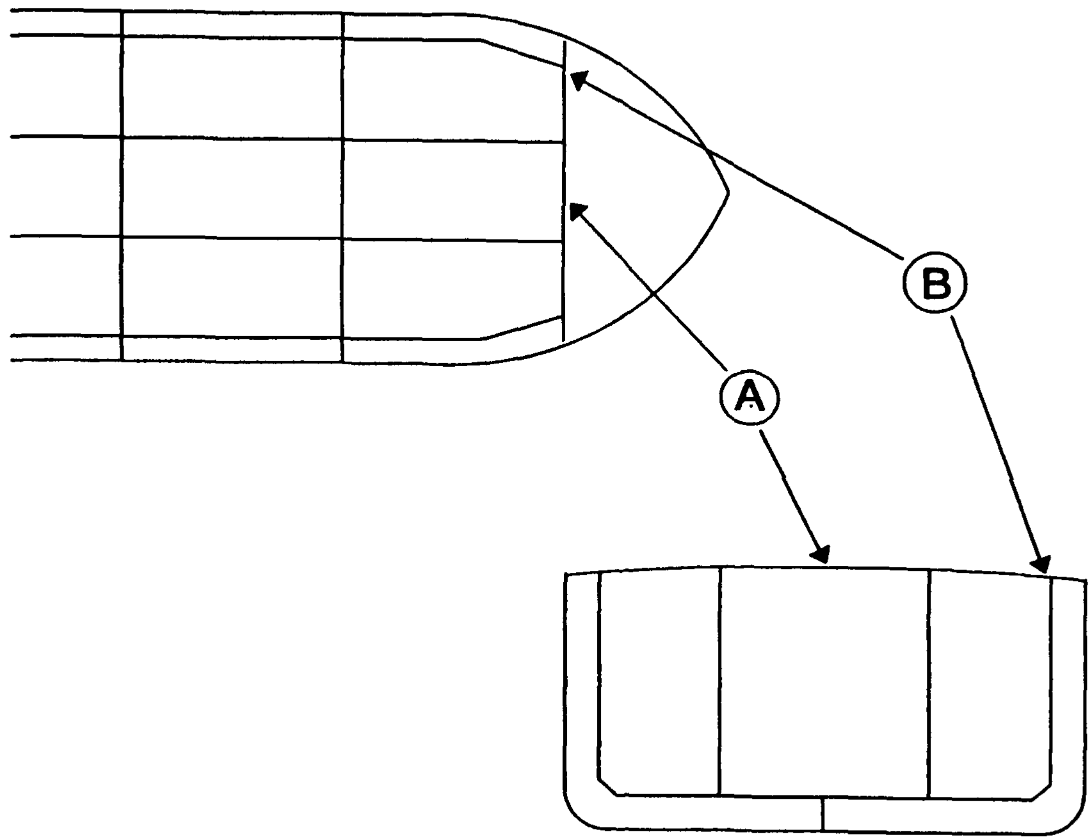
Critical Load Cases

	Full	Ballast	Partial 33%	Partial 50%	Partial 67%	Max. Static Hog	Max. Static Sag
VBM (Sag)	√		√				√
VBM (Hog)		√			√	√	
Vert. Accel.	√	√	√		√	√	
Lateral Accel.			√				
Roll		√	√	√	√	√	√

Load Components

- External Wave Pressure
- Internal Tank Pressure
- Inertial Loads due to Acceleration
- Hull Girder Shear Forces and Bending Moments

Representative Locations of Acceleration



APPLICATIONS AND BENEFITS

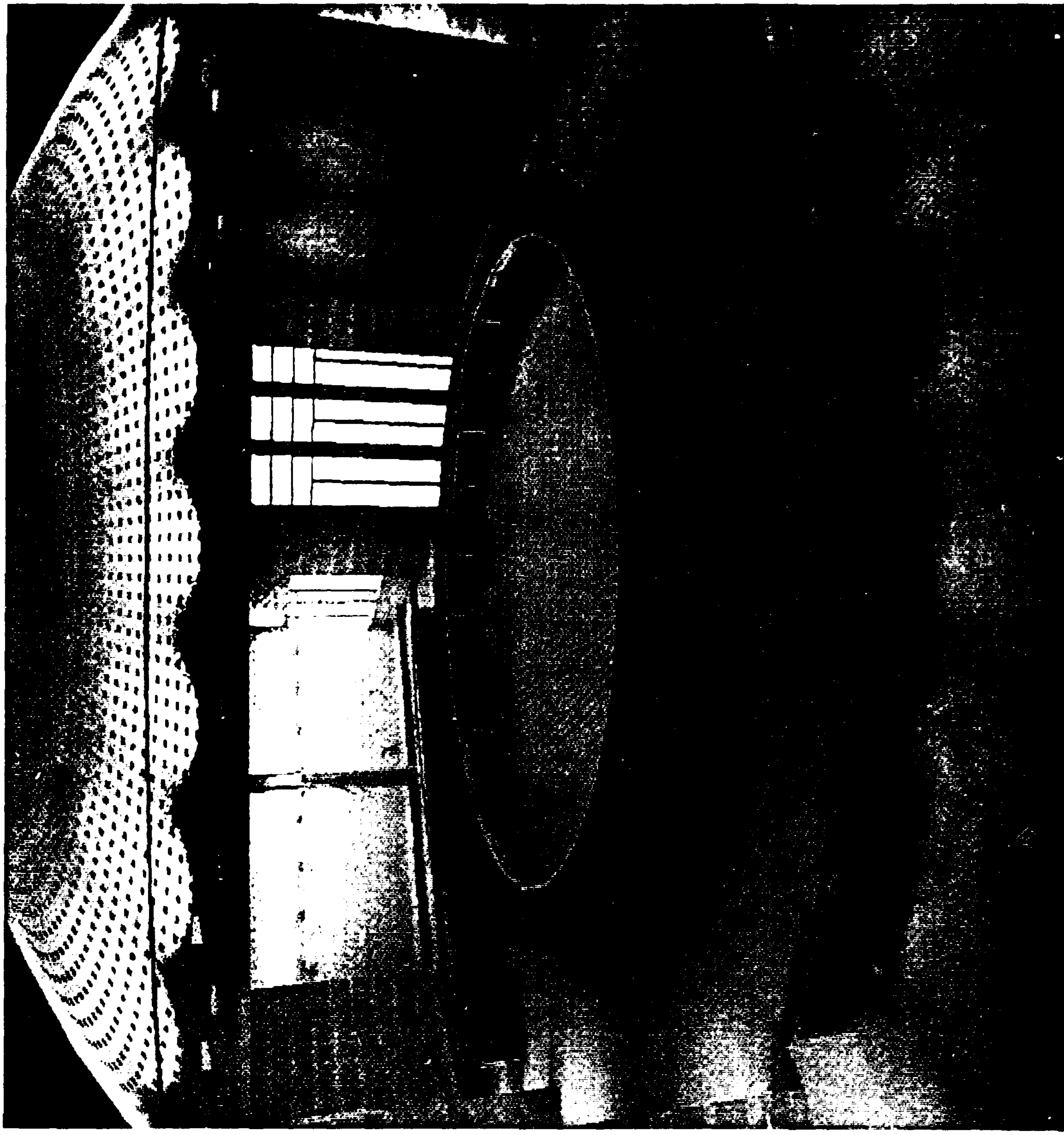
- VIRTUAL PROTOTYPING :
  - reduce or eliminate the need for costly and time consuming physical prototypes
  - rapid virtual prototyping allows for fast creation and modification of prototype

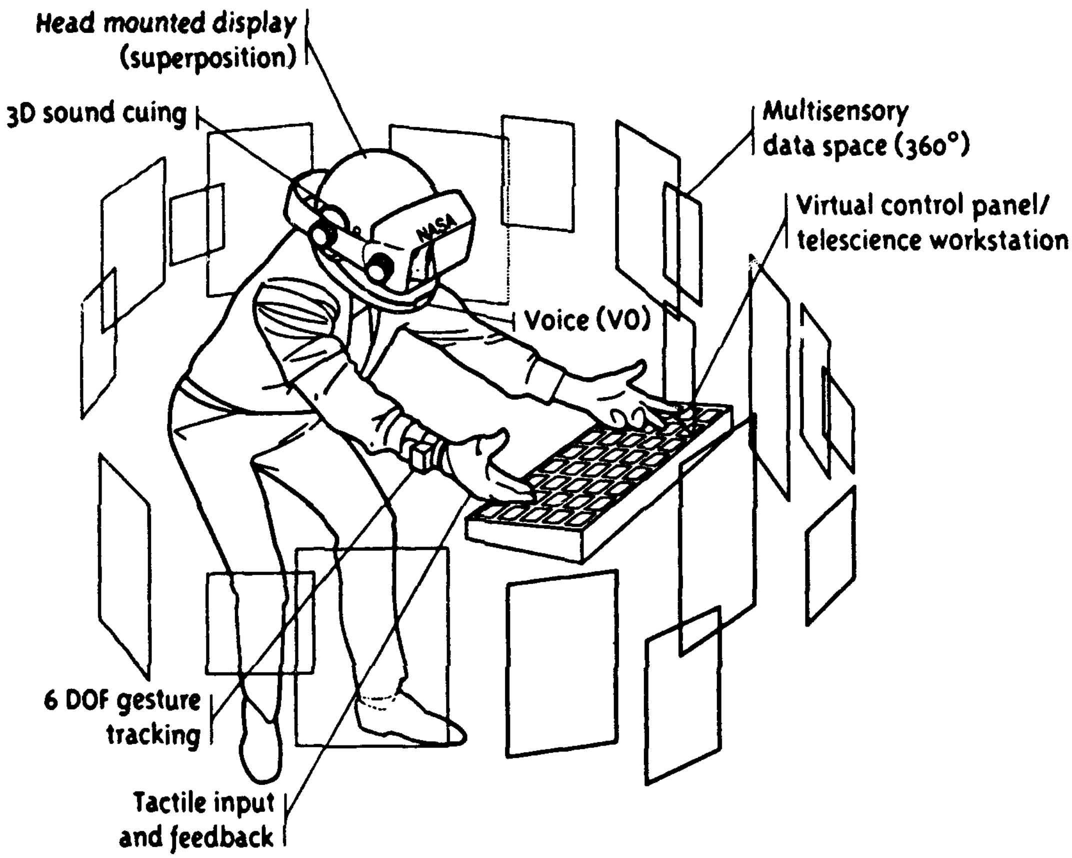
- ENGINEERING ANALYSIS :
  Integration of analysis results (e.g., CFD, FEM) with virtual prototypes

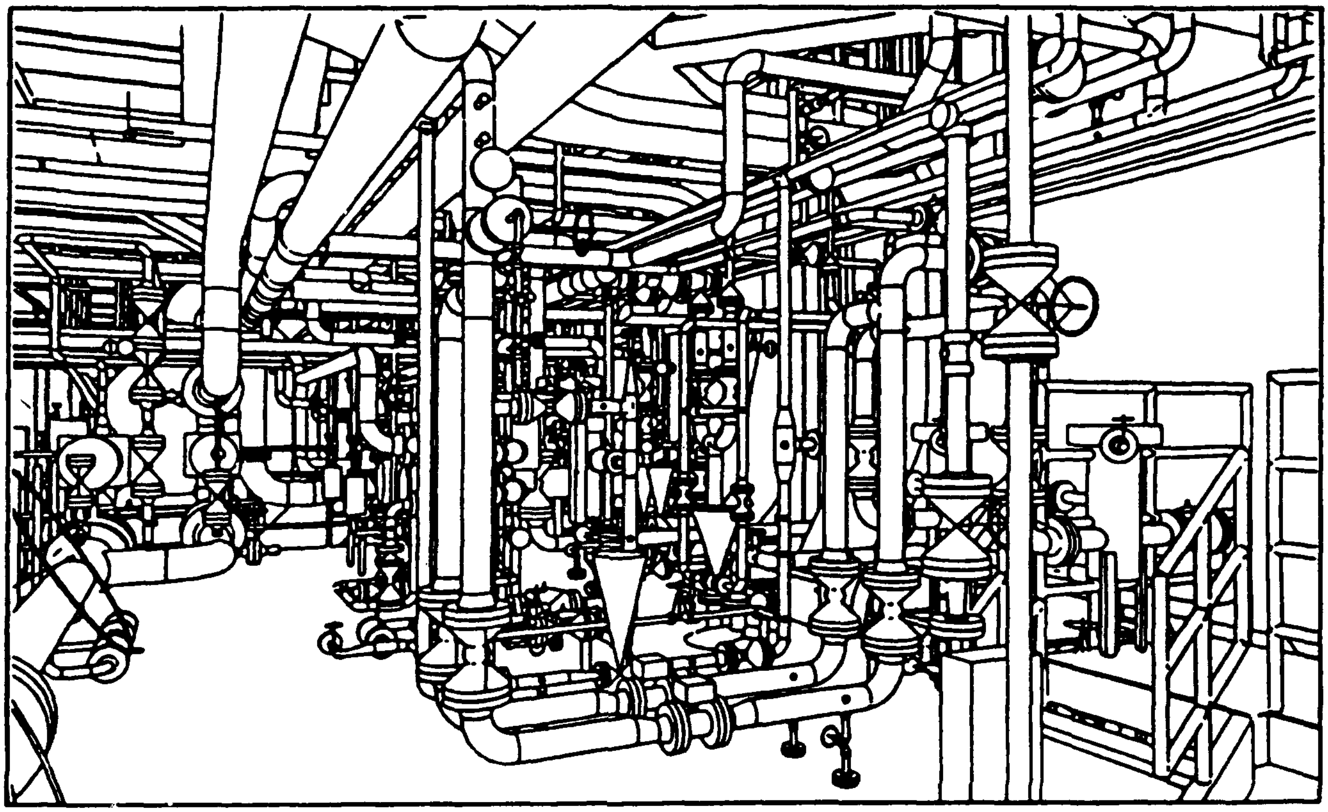
- OPERATIONAL SIMULATIONS :
  - direct involvement of humans for ergonomic, human factors, and performance studies
  - simulation of assembly, production, and maintenance tasks reveal problems at an early stage of the design process
  - training for operation, maintenance, safety

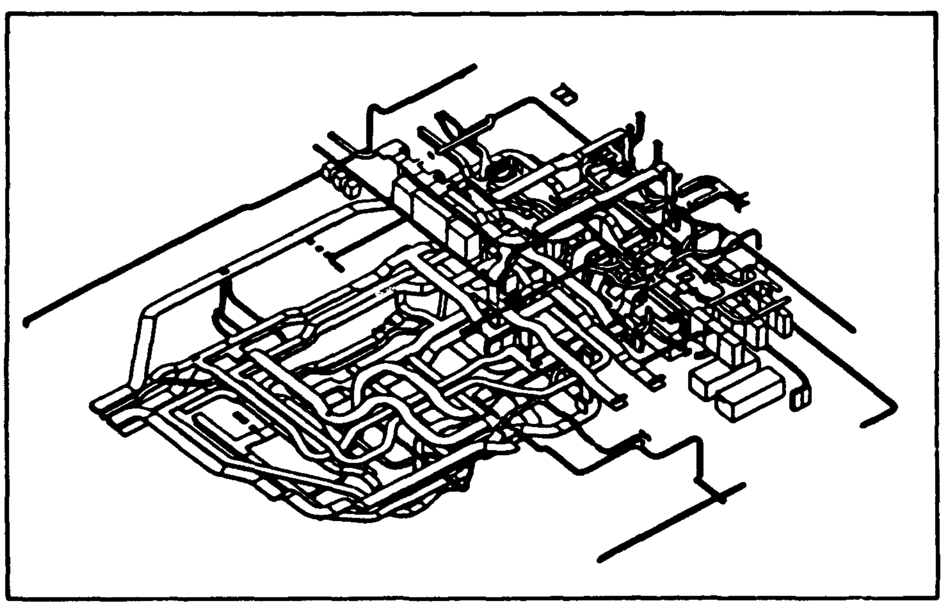
- CONCURRENT ENGINEERING :
  - explore all aspects of a design or a process by an interdisciplinary engineering team
  - shared virtual environments allow for participation from remote locations
  - VR as an integrating tool for : design - engineering - analysis - production planning - manufacturing - marketing & sales - maintenance - training

- OVERALL BENEFITS :
  savings in cost - savings in time - reduced design cycle - improved market response - better product




Head mounted display
(superposition)
3D sound cuing
Multisensory
data space (360°)
Virtual control panel/
telescience workstation
Voice (VO)
6 DOF gesture
tracking
Tactile input
and feedback



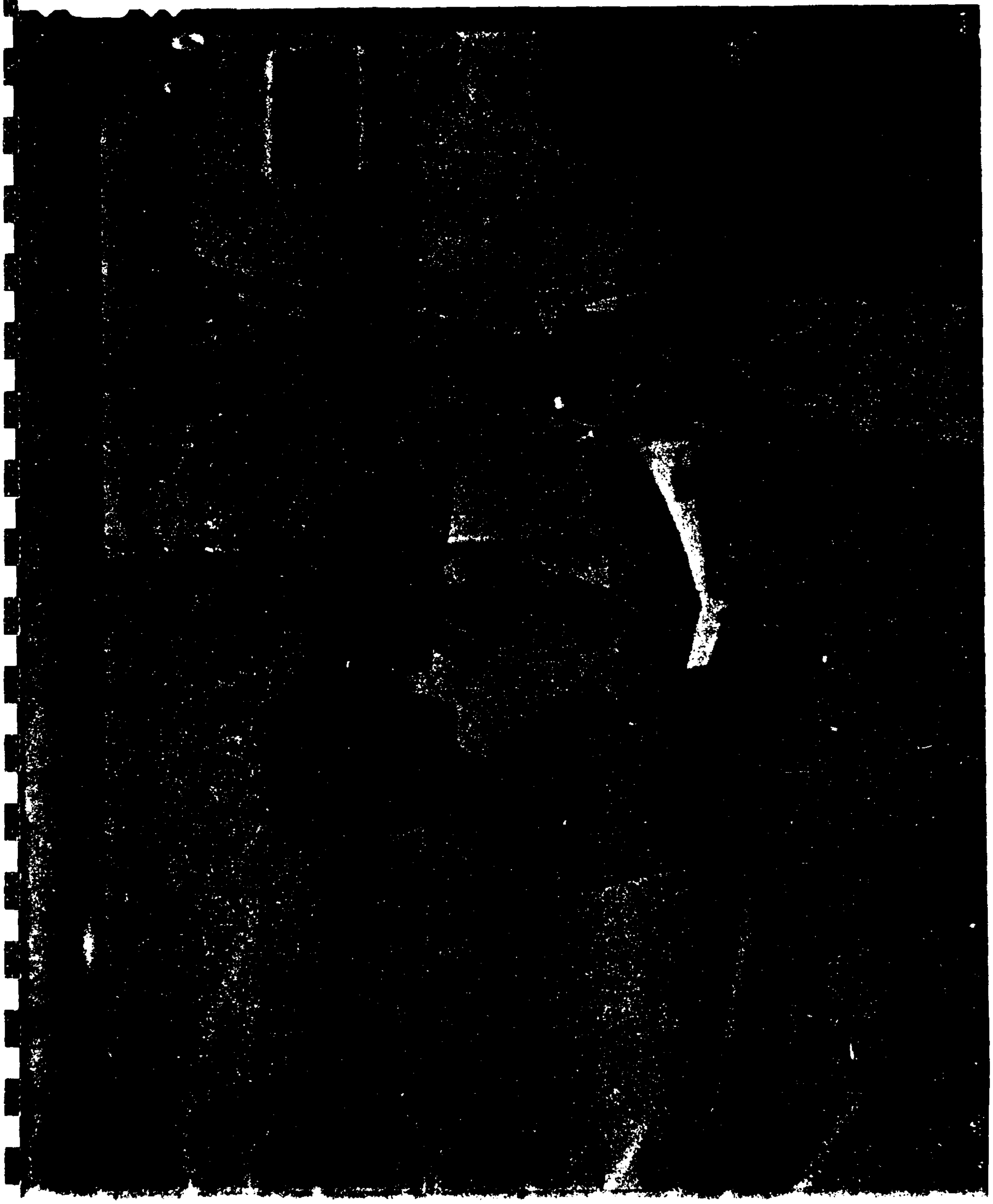


THE IMMERSIVE EXPERIENCE OF VR

- Convincing Illusion of Being Fully Immersed in an Artificial Three-Dimensional World
- Depth Perception through Stereo Viewing
- Full Look-Around & Walk-Around Capability
- Full Scale Representation of Virtual World
- Realistic Interactions with Virtual Objects
- Strong Sense of Realism and Spatial Perception

ASSETS FOR DESIGN AND MANUFACTURING

- Optimal Analysis Tool for Spatial Problems Involving Complex Three-Dimensional Geometry (arrangements, mechanical systems, abstract systems)
- Realistic Integration of Humans with Virtual World (especially effective if humans are part of a system)
- Optimal Communication & Demonstration Tool



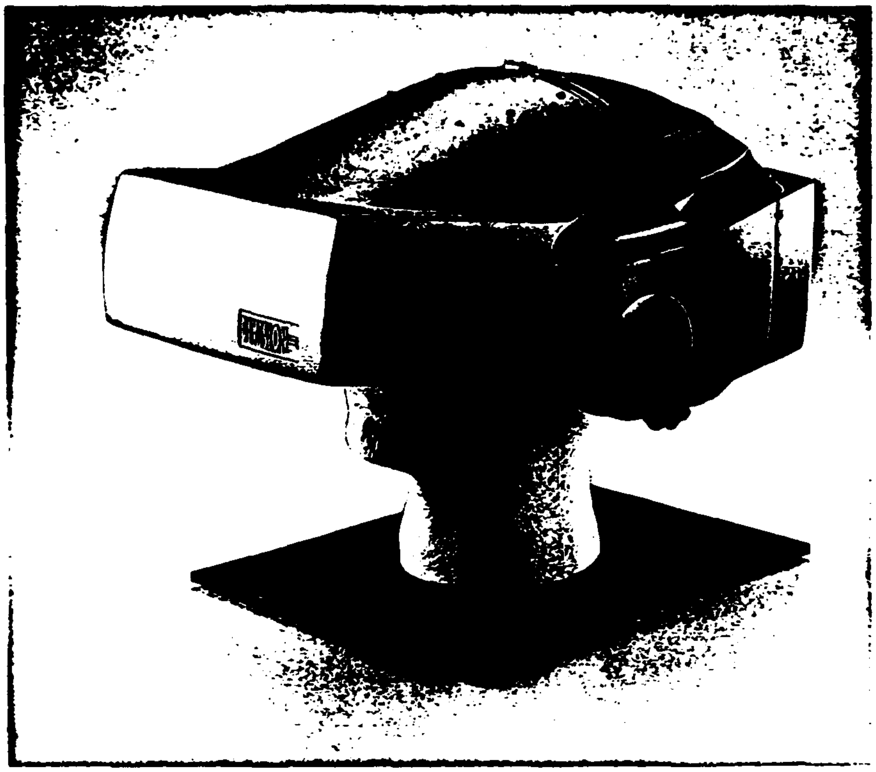


Figure 1. A Head-mounted display (HMD) provides fully immersive virtual reality.

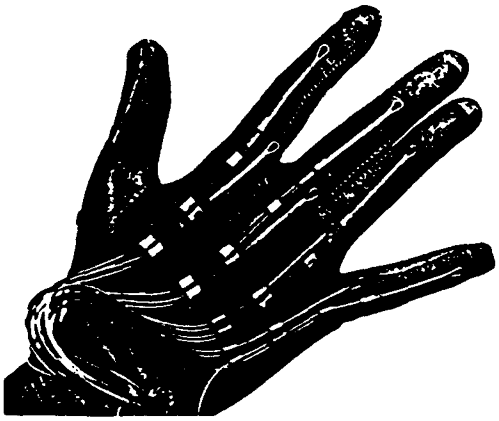
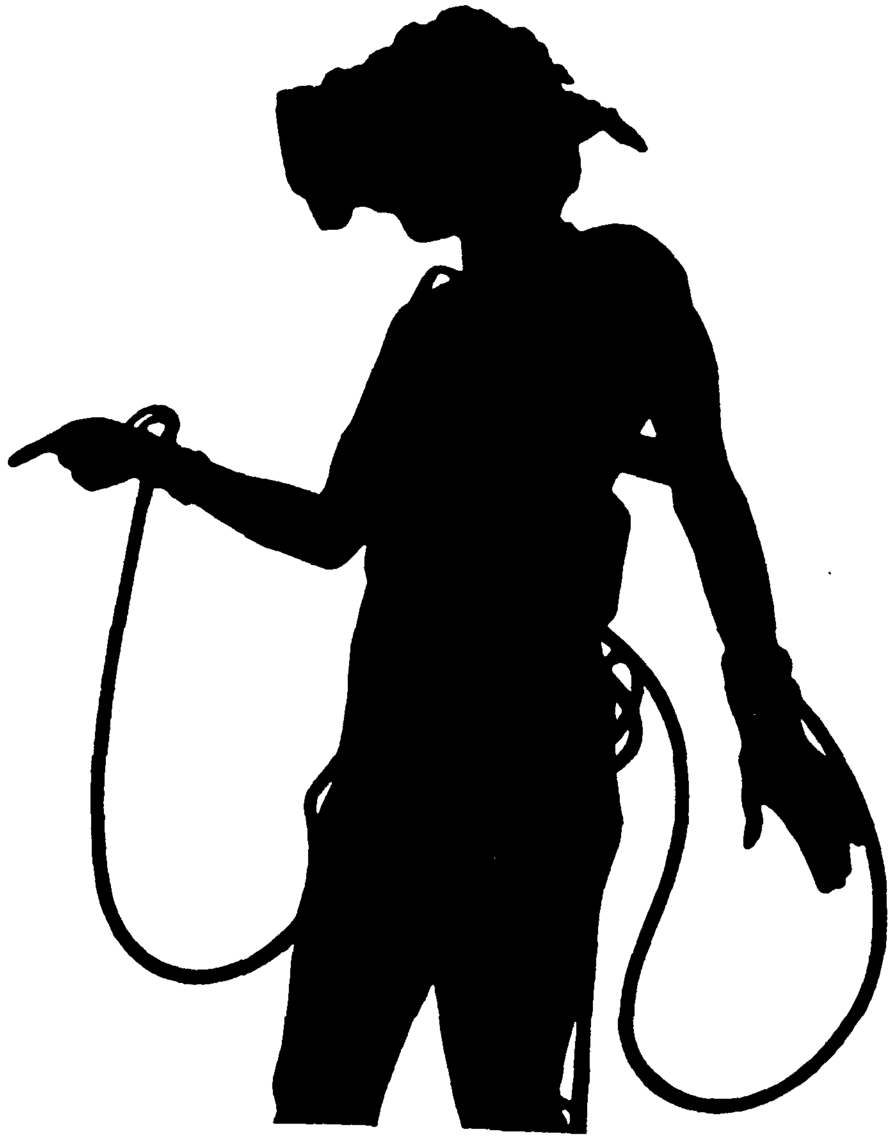
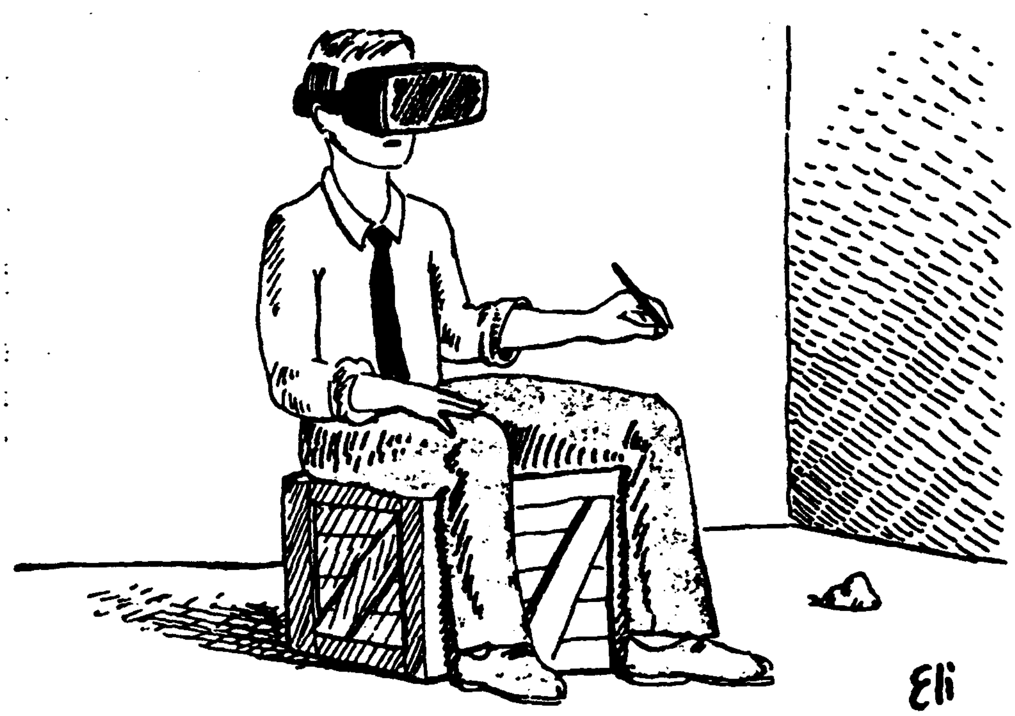


Figure 2. Data glove with flexion sensors and tracking receiver.

THE ENABLING TECHNOLOGIES OF VR

- **Head-Mounted Display (HMD)**
- **Motion Tracking System**
- **Image Generation System**
- **Interactive Input Devices (Gloves, Suits)**
- **Tactile and Forced Feedback**
- **Additional Component Technologies**
  - Eye Tracking Systems
  - Telepresence Technologies
  - Directional 3D Sound
  - Voice Recognition / Speech Synthesis
- **Alternative Display Technologies**
  - Head-Coupled Display (HCD)
  - See-Through HMD/HCD
  - Shutter Glasses
  - Large Screen Projection
  - Retinal Laser Display





The virtual office

Klaus-Peter Beier

University of Michigan

VIRTUAL REALITY (VR)
VIRTUAL ENVIRONMENTS (VE)

IN DESIGN AND MANUFACTURING

- The Enabling Technologies of Immersive VR
- Applications in Design and Manufacturing
- Virtual Prototyping (case study: automotive interiors)
- Augmented Reality in Assembly and Maintenance
- The University of Michigan Virtual Reality Laboratory

CMS-SSC STRUCTURAL RELIABILITY THRUST

OTHER RELIABILITY PROJECTS

SSC 363 - 1992
Uncertainties in Estimating Loads and Load Effects on Marine Structures
Estratos Nikoladis
Developed estimates of bias and uncertainty in loads and load effects

SSC 371 - 1993
Establishment of a Uniform Format for Data Reporting of Structural Material Properties for Reliability Analysis
Fleet Technology Limited
Developed a standard format so that the properties of materials will be available in a probabilistic format

SR - 1338
Uncertainty in Strength Models for Marine Structures
Owen Hughes
Objective - Quantify bias and uncertainty in structural strength formulations in order to evaluate safety margins and derive design criteria.

SR-1344
Assessment of Reliability of Existing Ship Structures
Alaa Mansour
Will estimate the reliability levels associated with important failure modes for several existing ships

CMS-SSC STRUCTURAL RELIABILITY THRUST

PHASE III - SR-1345
Probability-based Ship Design: Implementation of Design Guidelines for Ships
Alaa Mansour
Project began in May 1994 with the objective of developing ship structural design procedures that are reliability-based.

PHASE IV - SR-1362 (New Phase)
Probability-based Design, Synthesis of the Reliability Thrust Area
Project to begin in FY 1994
Have a group of experts produce a summary of the state of the art

PHASE V (Old Phase IV)
Probability-based Design: Novel Hull Forms and Environments
Will extend previous work to unconventional hull forms and to unusual load situations

CMS-SSC STRUCTURAL RELIABILITY THRUST

1983 DESIGN, INSPECTION AND REDUNDANCY SYMPOSIUM
Recommended a program of several projects to determine and unify reliability of marine structural systems

June 1987 ad hoc Reliability Committee
Developed Four-Phase Reliability Thrust

SSC-351 - 1990
Alaa E. Mansour and Paul F. Wirsching
A tutorial on structural reliability theory
The basis for SSC reliability work

PHASE I - SSC 368 - 1993
Probability Based Ship Design Procedures - A Demonstration
Alaa Mansour
Demonstrated the use of probability-based design in the analysis of a tanker and a combatant ship

PHASE II - SSC 373 - 1994
Probability-based Ship Design: Loads and Load Combinations
Alaa Mansour
Developed standard loads necessary for a probability-based design

COMMITTEE ON MARINE STRUCTURES AND SHIP STRUCTURE COMMITTEE

STRUCTURAL RELIABILITY THRUST

Robert A. Sielski
Marine Board
National Research Council

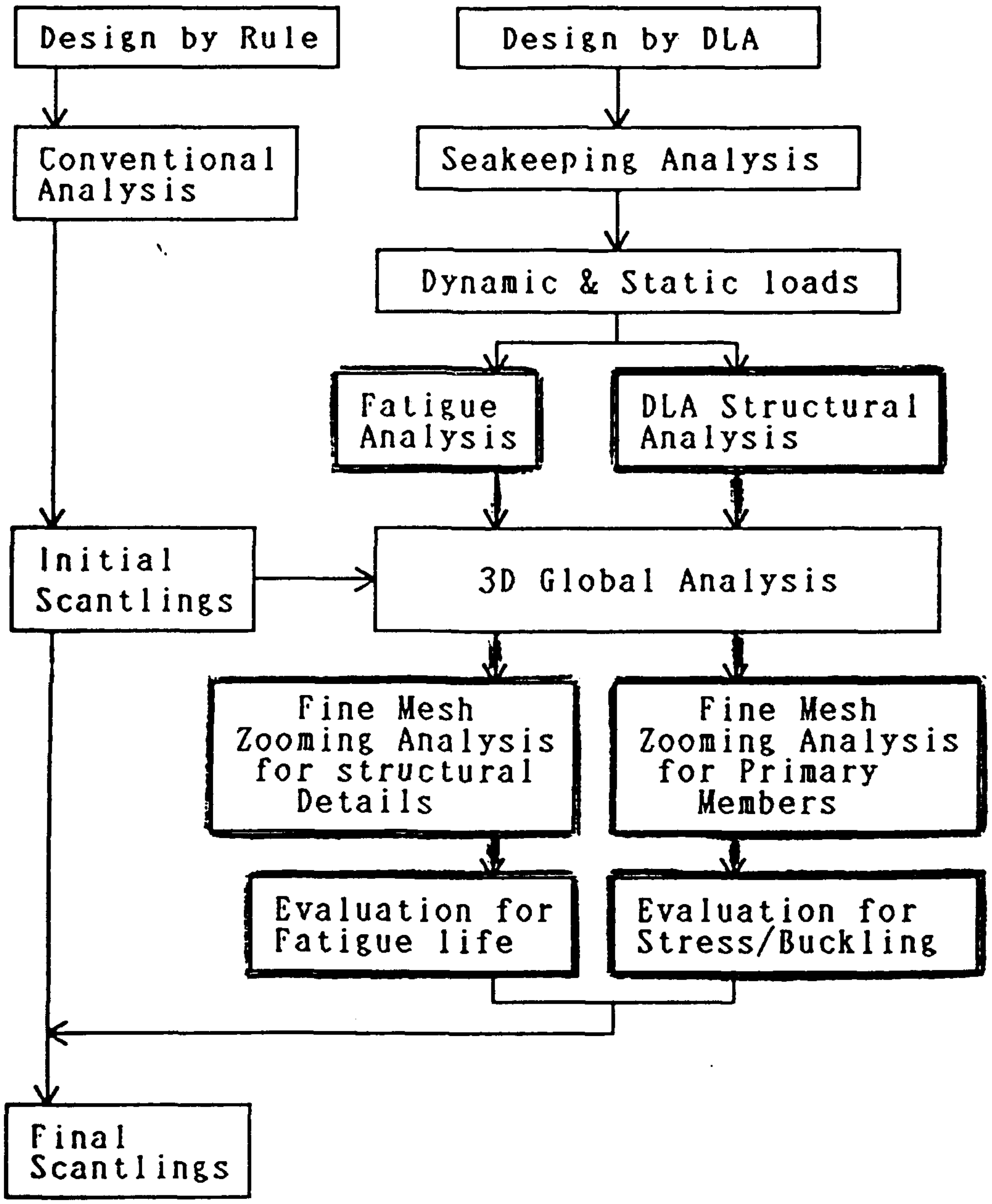
July 7, 1994

Summary

- Critical Cargo Loading Condition
  - Partial Load
- Critical Load Cases
  - Max. Roll and Vert. Acceleration
- Scantling Increases in Local Structural Members
  - Web Frame, Hor. Girders
- Steel Weight Increase is Approximately 100-200 tons
- Existing Rules are Basically Adequate
- Stronger, More Robust, and Longer Life Vessels

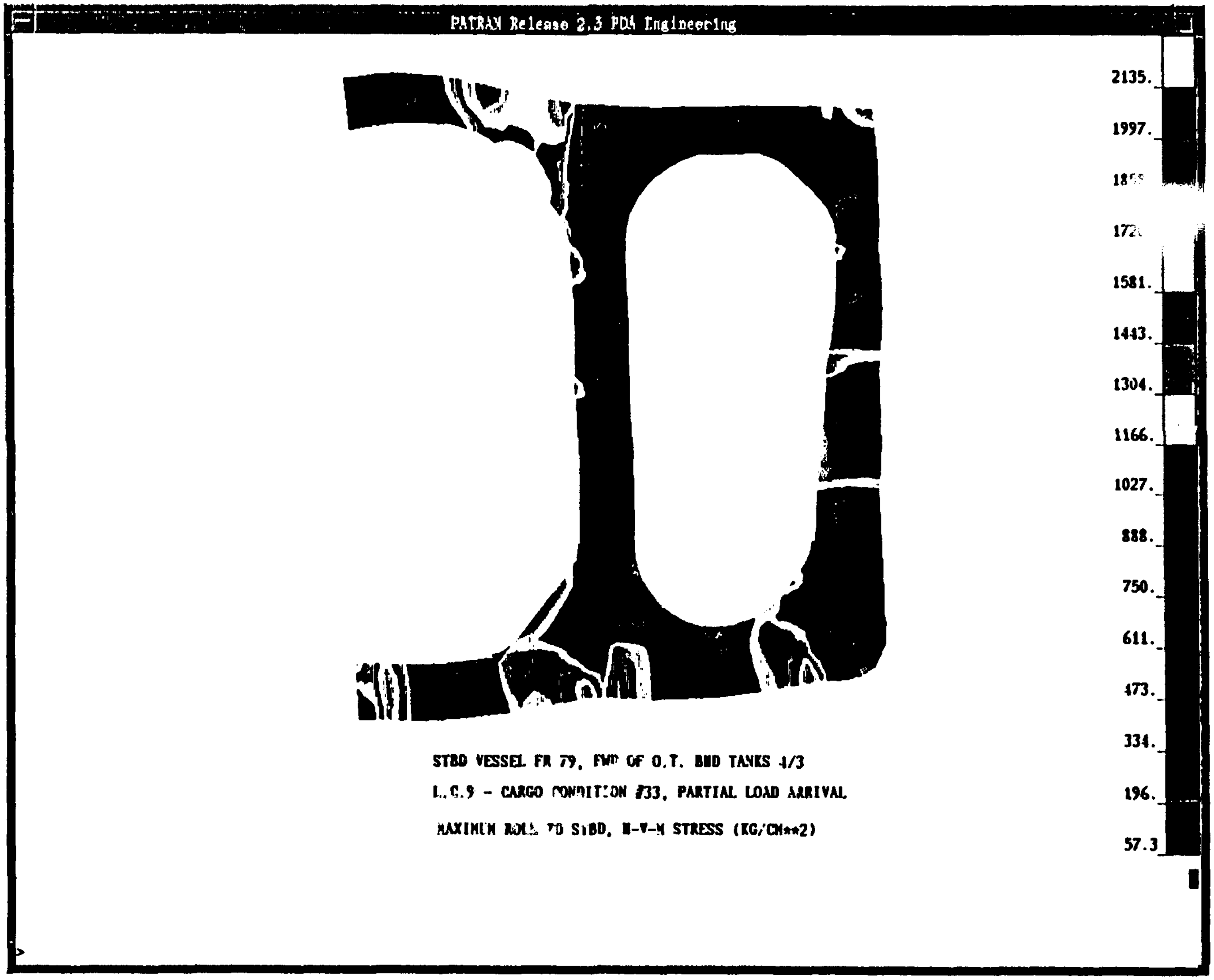
Acceptance Criteria

- Yielding Criteria
  - Henckey-von Mises Stress
  - 95% of Yield Strength
- Buckling Criteria
  - Plate Panels and Supporting Members
  - Elastic Buckling Criteria
- Fatigue Criteria
  - Miner's Cumulative Damage
  - UK DEN S-N Data
  - 20-year Life

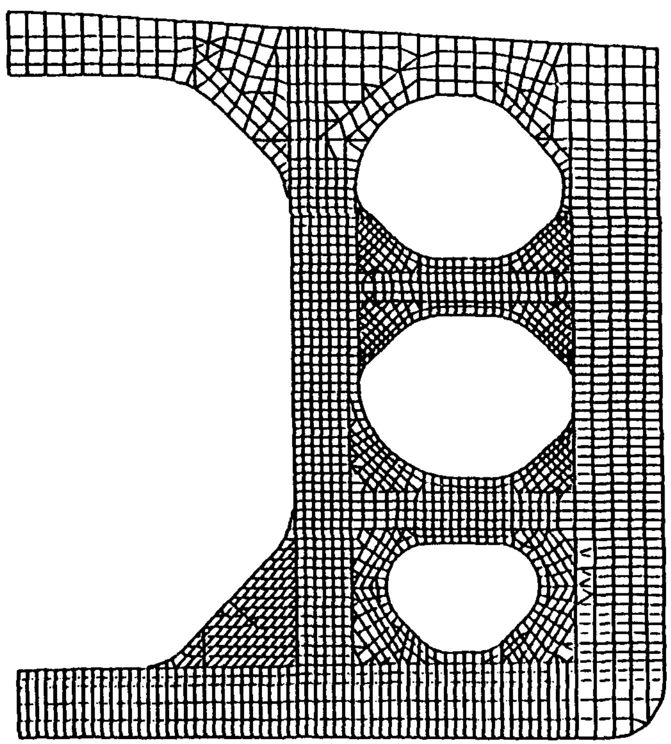


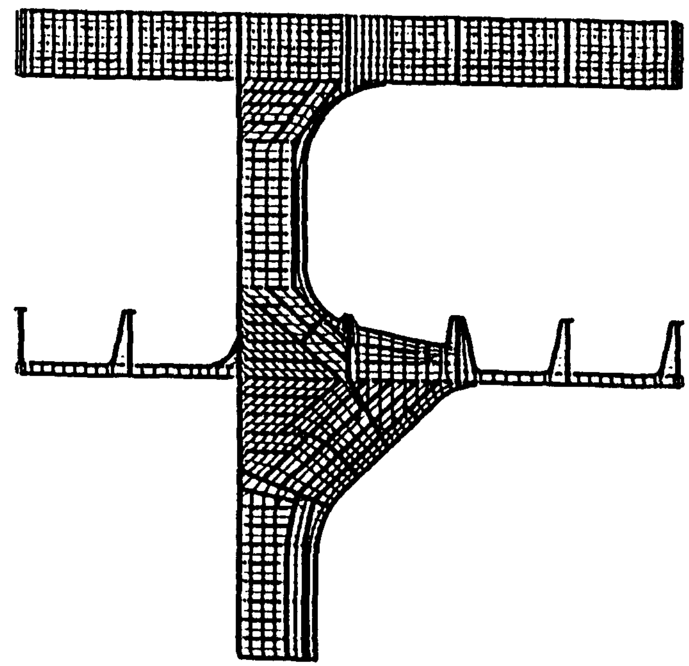
Design Procedure Using Dynamic Load Approach

Typical Fine Mesh FEM Result for Web Frame



A Typical Fine Mesh FEM Model of a Tanker





Fine Mesh FEM Analysis of Local Structure

- Structural Members required for Analysis
- Boundary Conditions from 3-D Analysis Result
- All Loads Applied to Fine Mesh Model

VIRTUAL PROTOTYPING

- Replace costly and time-consuming physical mockups
- Rapid Virtual Prototyping: fast creation and modification
- Create virtual prototype from existing CAD/CAM data
- Apply rendering algorithms, lighting models, texture mapping, and other techniques for realistic appearance
- Realistic interactions with prototype via data glove, etc.
- Combine virtual display with physical elements if correct forced feedback is needed (e.g., simplified seating buck)
- Examples of Extended Functionality :
    - employ transparent display techniques for inspection of hidden components
    - use prototype already for the analysis of incomplete designs (e.g., with parts floating in space)
    - superimpose design alternatives for comparison
    - allow for interactive design modifications with immediate feedback
- Usage of Virtual Prototypes :
    - Design analysis (clearances, packaging efficiency, connectivity, motion characteristics, collision, ...)
    - Human factors studies (visibility, reachability, accessibility, comfort, human performance, appeal, ...)
    - Base for other VR applications (engineering analysis, operational simulations, concurrent engineering, shared virtual environments, training, marketing, ..)

CREATION OF VIRTUAL PROTOTYPES

- **GEOMETRY DEFINITION :**

  CAD/CAM Model : mathematical description through surface modeling, solid modeling, and related methods

  Virtual Model : approximation by computer graphics primitives (mainly polygons and polygon-meshes)

- **PROCESS :**

  Access CAD/CAM database and extract geometry

  Approximate boundaries by polygons (tessellation)

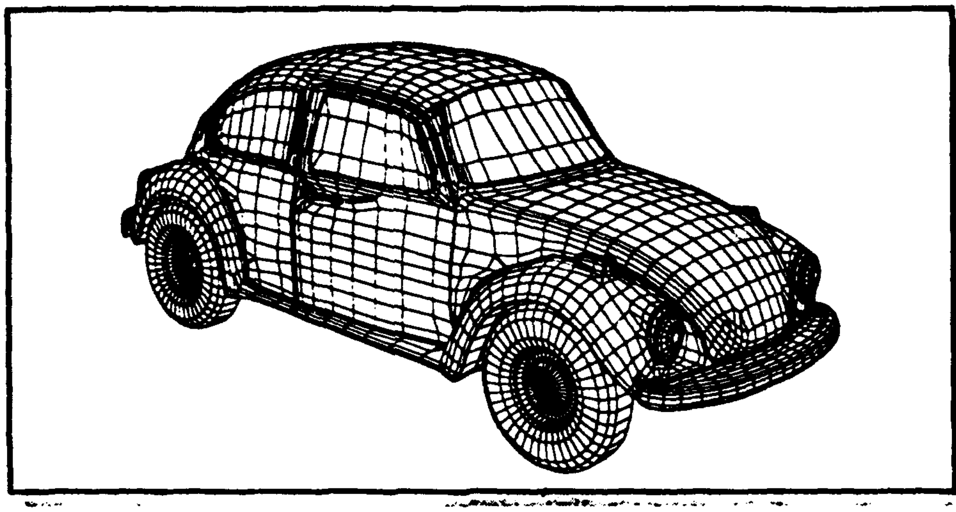
  Simplify geometry : reduce number of polygons for graphics performance, decide on level of detail

  Edit geometry : identify and remove unnecessary geometry, correct faulty geometry, incorporate textures

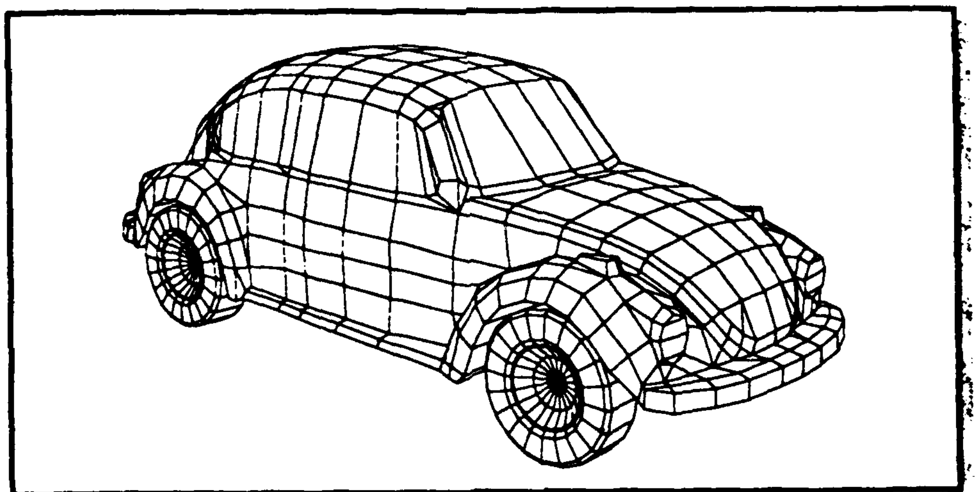
  Define display characteristics : color, reflection characteristics (material properties), transparency, lighting configuration, rendering method, ...

  Model the behavior of prototype (define interactive feedback mechanisms and trigger events, dynamic and other responses, motion restrictions, etc.)

  Calibrate virtual environment with user, data glove and physical elements



73 VW Super Beetle-H — 10364 Vertices, 10514 Polygons



73 VW Super Beetle-L — 1520 Vertices, 1602 Polygons

Polygonal representations with different levels of accuracy

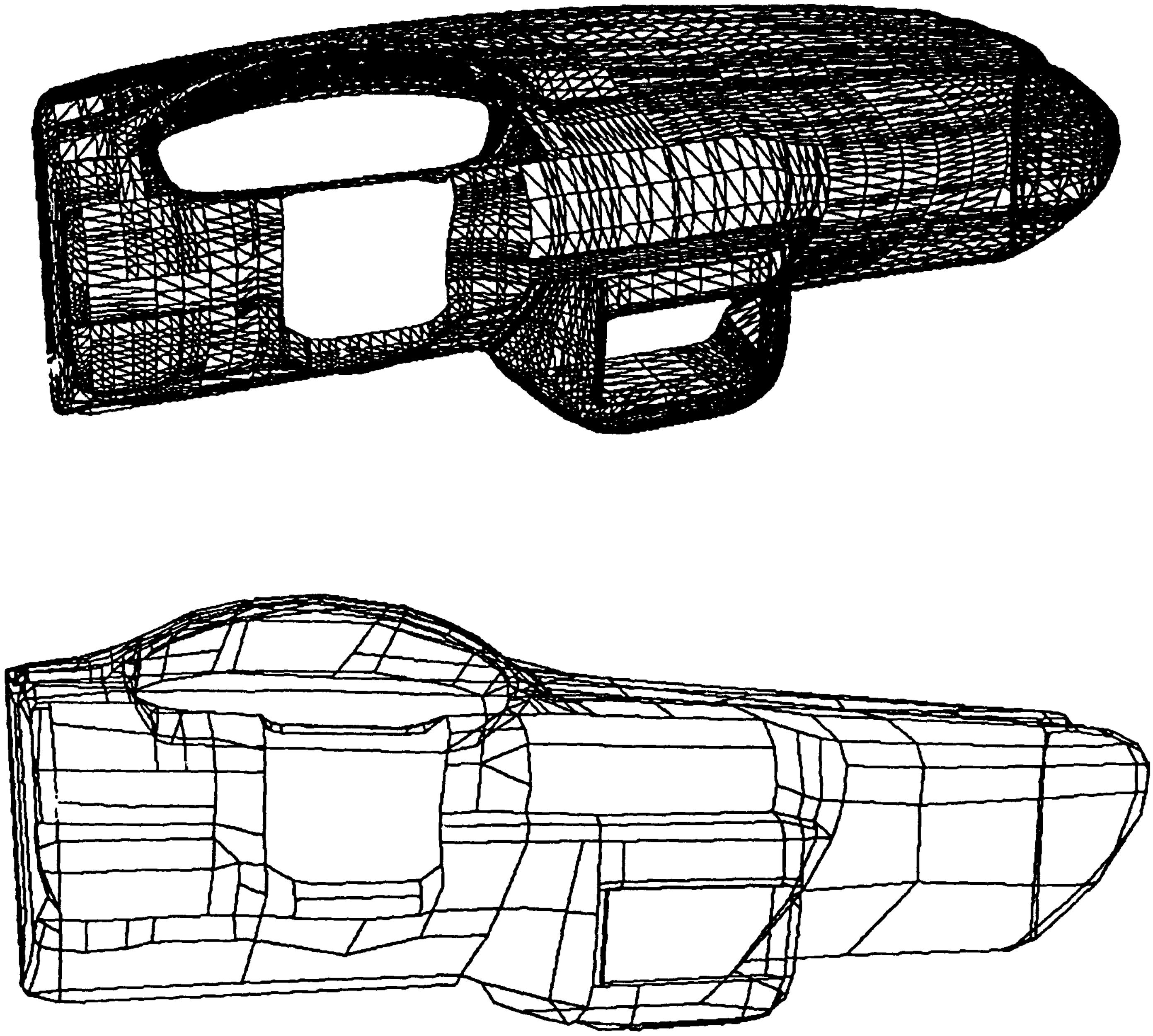
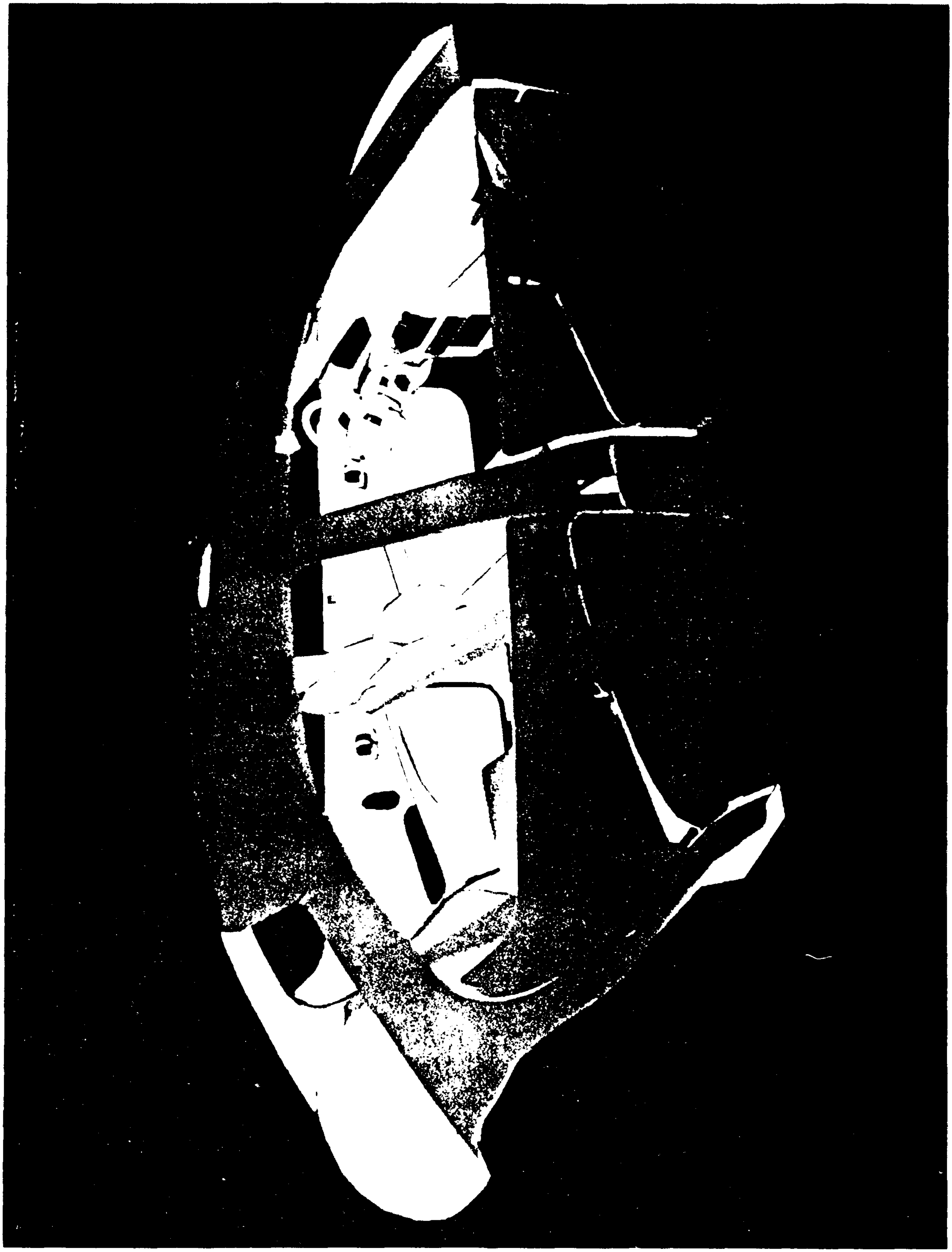
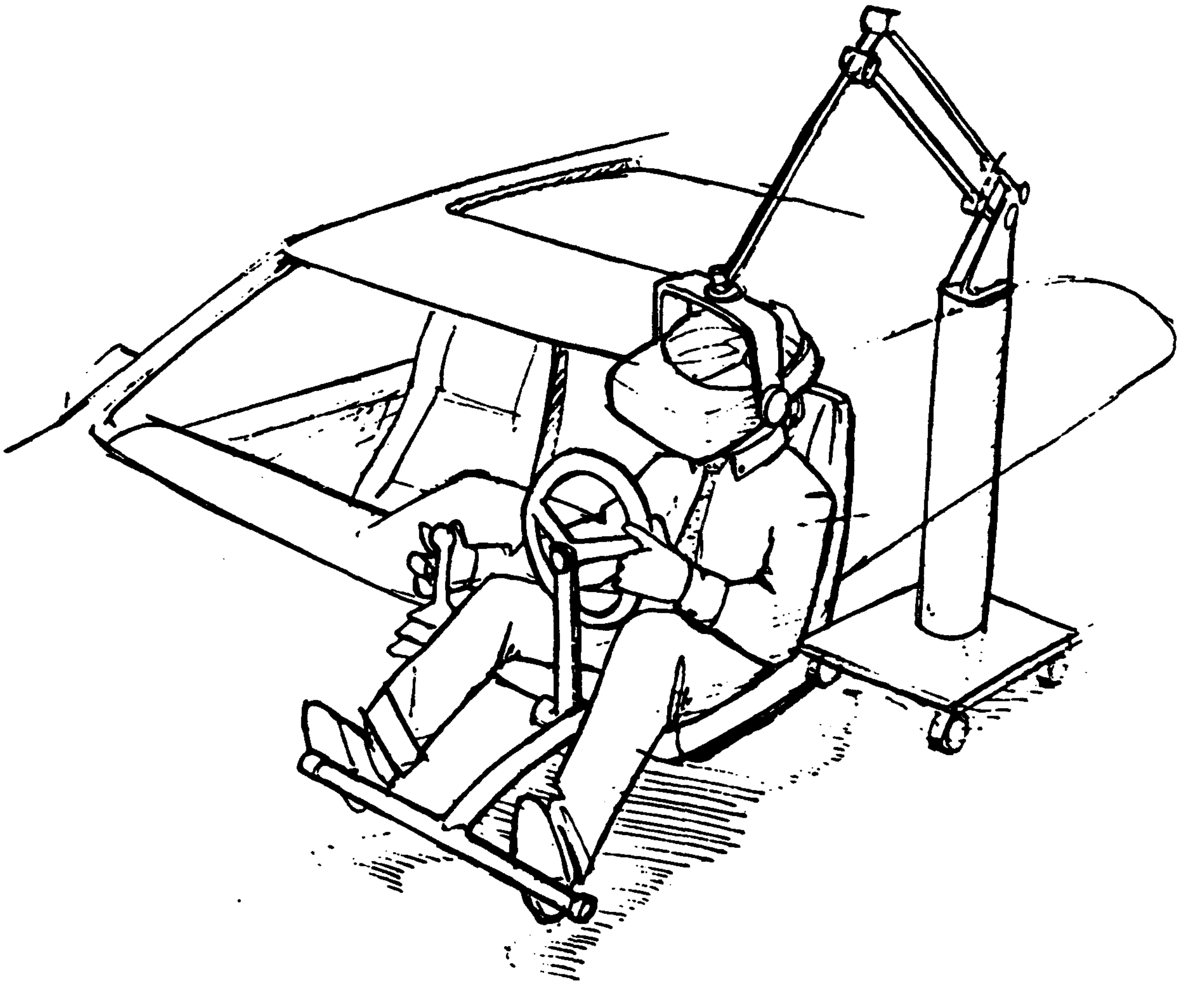
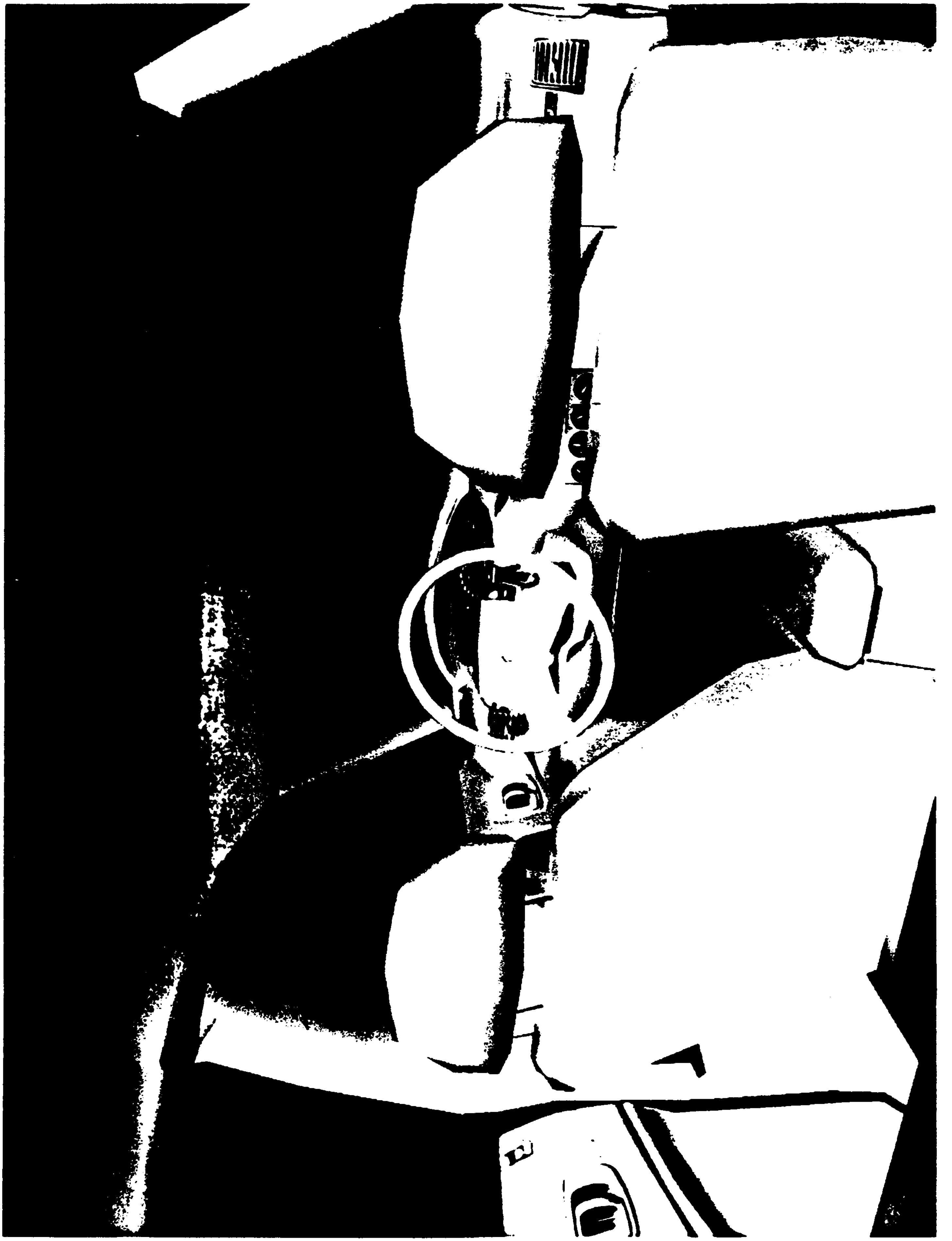


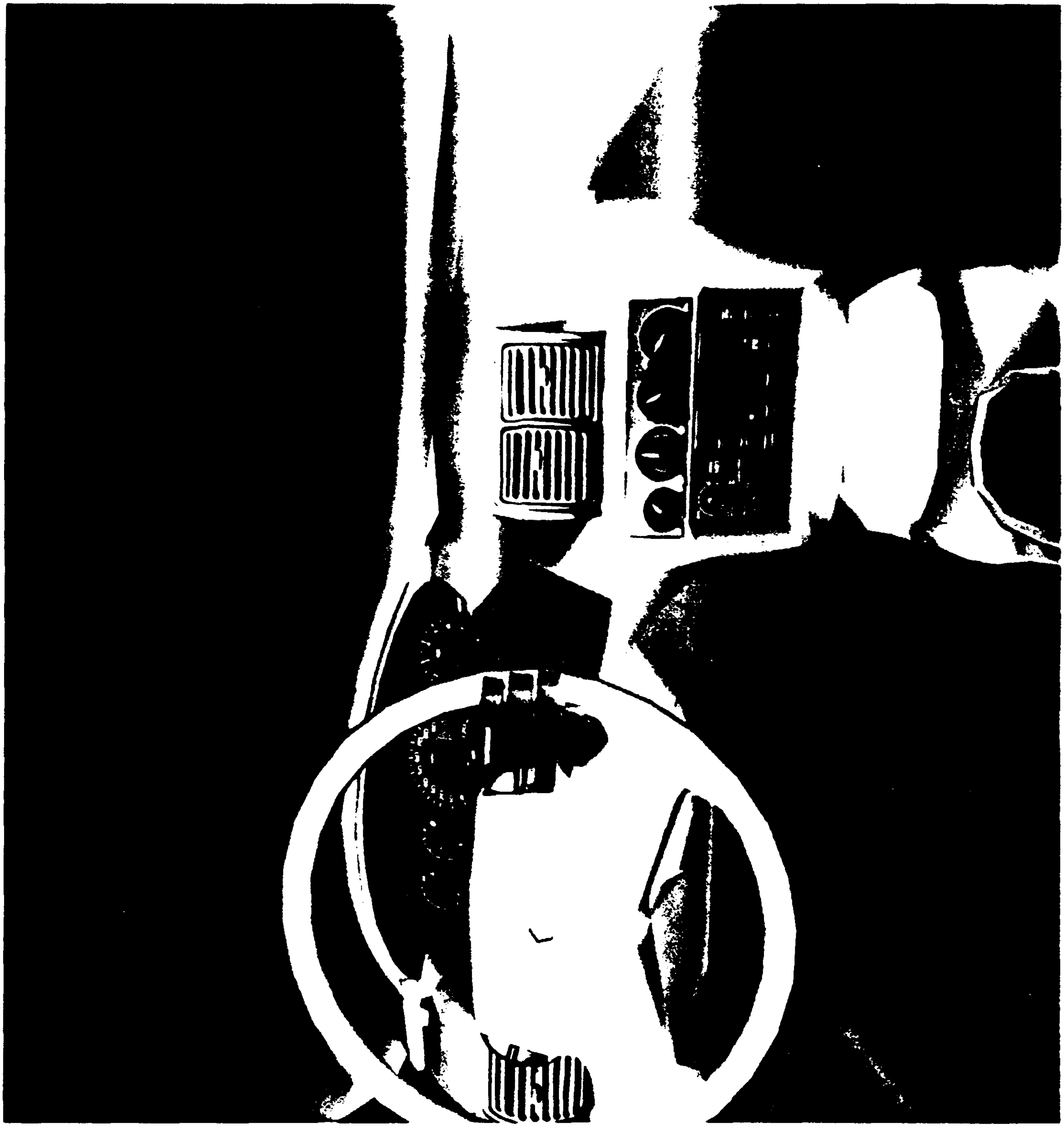
Figure 10. Polygonal approximation of a panel derived from a CAD/CAM model (top) and simplified VR geometry (bottom).

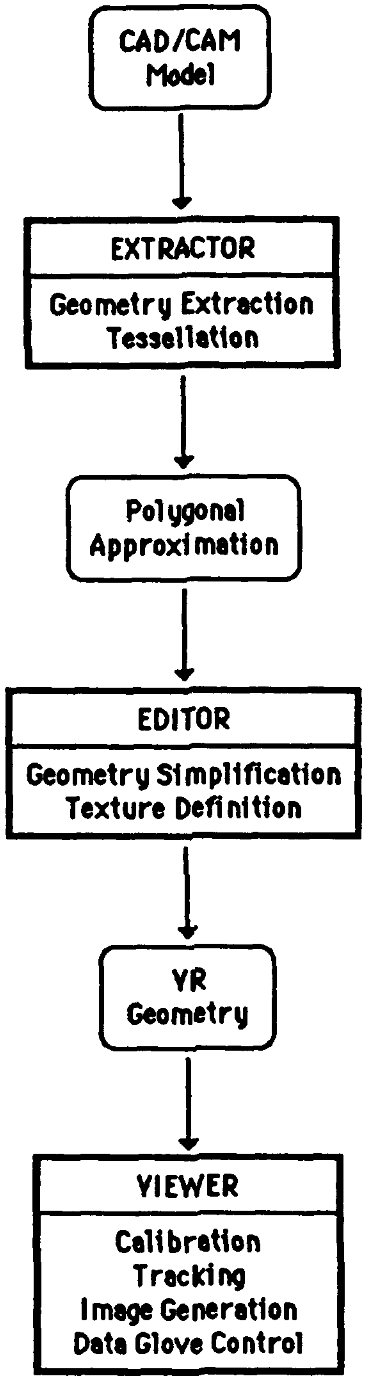

editSVR
File
Edit
Selection
Editors
Manips
Lights
Options
Rotx
Roty
Zoom
Dolly
manipulator Mode (normal)











**The process of generating
a virtual prototype from a CAD/CAM model**

VIRTUAL REALITY LABORATORY

THE UNIVERSITY OF MICHIGAN - COLLEGE OF ENGINEERING
Department of Naval Architecture and Marine Engineering

Created: April 1993 through an initial grant from Chrysler with cost sharing by The University of Michigan

Director: Klaus-Peter Beier

Focus: VR as an integrating concurrent engineering tool for :
design - engineering - analysis - production planning - manufacturing - marketing & sales - maintenance

Goals:

- Advance VR technology (cross the threshold of usability)
- Develop methods, algorithms, and software concepts
- Prove usefulness through demonstration projects
- Assist industry with the introduction of VR

Laboratory Equipment :

- SGI Onyx with 2 processors and VTX graphics (currently upgrading to 4 processors and Reality Engine graphics)
- Boom2C from Fakespace (currently upgrading to Color Boom3C)
- DataGlove from VPL, Isotrack from Polhemus
- IBM-RS/6000 with CAD/CAM system CATIA
- SGI Indigo2/Extreme, SGI Iris, SGI Indy, HP and Sun workstations, Macintoshes, Scanner, Video production equipment, others, ...

VIRTUAL REALITY LABORATORY

ONGOING PROJECTS :

- **Virtual Prototyping of Automotive Interiors**
  - for design analysis and human factors studies
  - sponsored by Chrysler Corporation

- **Virtual Environments as an Analysis Tool in Computational Fluid Dynamics and Crash Simulations**
  - focus on high performance computing applications
  - sponsored by DoE (CRADA)
  - in cooperation with five National Laboratories and with Chrysler, Ford, and General Motors

- **Virtual Interior Arrangements for Sailing and Motor Yachts**
  Department of Naval Architecture and Marine Engineering

- **Virtual Model: Integrated Technology Instruction Center**
  future home of the Virtual Reality Laboratory, under construction

IN PREPARATION :

- **Augmented Reality in Simulation-Based Design**
  - applications in design, assembly, and maintenance
  - sponsored by ARPA
  - to be integrated with Chrysler project on virtual interiors

- **Shared Virtual Environments**
  - simultaneous immersion of users on both sides of the Atlantic
  - in cooperation with the Computer Graphics Center (ZGDV) Darmstadt, Germany, Ford/US, and Ford/Europe

- **Virtual Diving**
  - virtual models of underwater terrain, structures, ship wrecks
  - training of divers and ROV operators (M-ROVER)
  - planning of underwater operations

ONR WORKSHOP

NONLINEAR SEA LOADS AND SHIP RESPONSE: A BASIS FOR SHIP STRUCTURAL DESIGN

The Role of Simulation in Ship Design: Some Cautionary Examples

presented by

Armin W. Troesch, PhD, PE

Department of Naval Architecture and Marine Engineering

The University of Michigan

Ann Arbor, Michigan

The University of Michigan

THE ROLE OF SIMULATION IN DESIGN

With regards to the new computer simulators...

"If you choose the right parameters, your simulated experiment will normally work just like the corresponding real-world experiment. (However,) there are some edges to this simulated world, and if you step over the simulation breaks down badly..." (Swaine, 1992)

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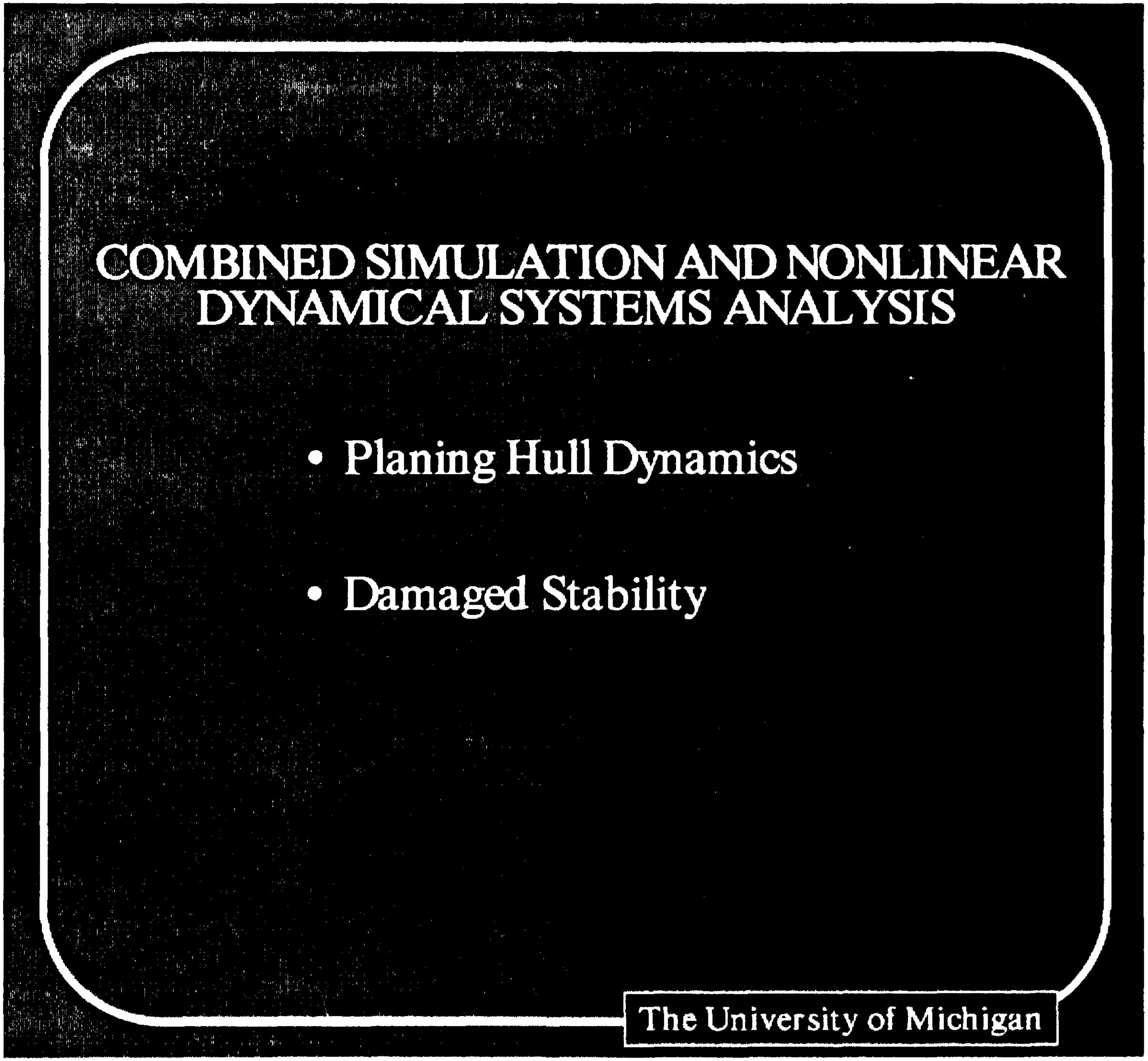
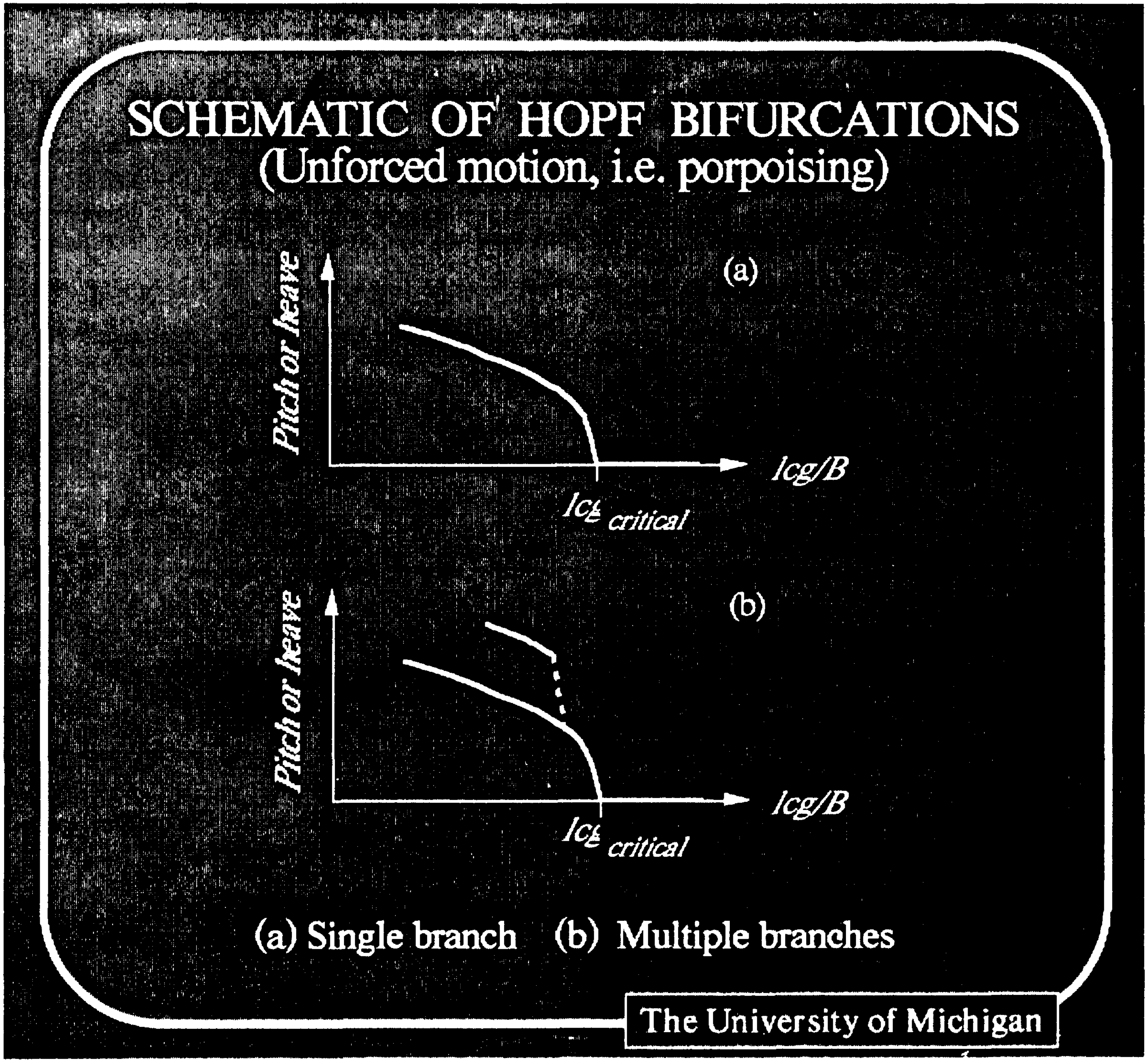
RATIONALE:

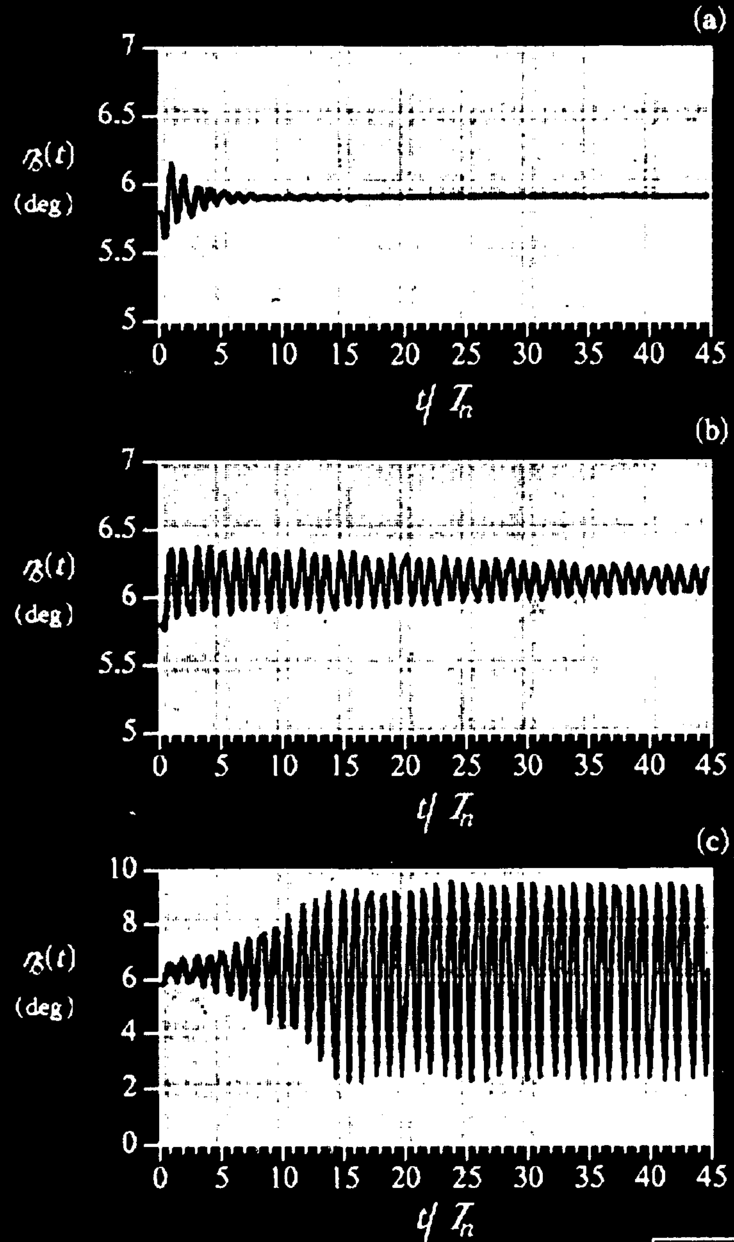
Identification of design parameters
that are critical to performance.

Example: "A Case Study of Dynamic Instability
in a Planing Hull",

Codega and Lewis. *Marine Tech.,* April, 1987


COMBINED SIMULATION AND NONLINEAR DYNAMICAL SYSTEMS ANALYSIS
• Planing Hull Dynamics
• Damaged Stability
The University of Michigan


SCHEMATIC OF HOPF BIFURCATIONS
(Unforced motion, i.e. porpoising)
(a)
Pitch or heave
lcg/B
lcg critical
(b)
Pitch or heave
lcg/B
lcg critical
(a) Single branch (b) Multiple branches
The University of Michigan

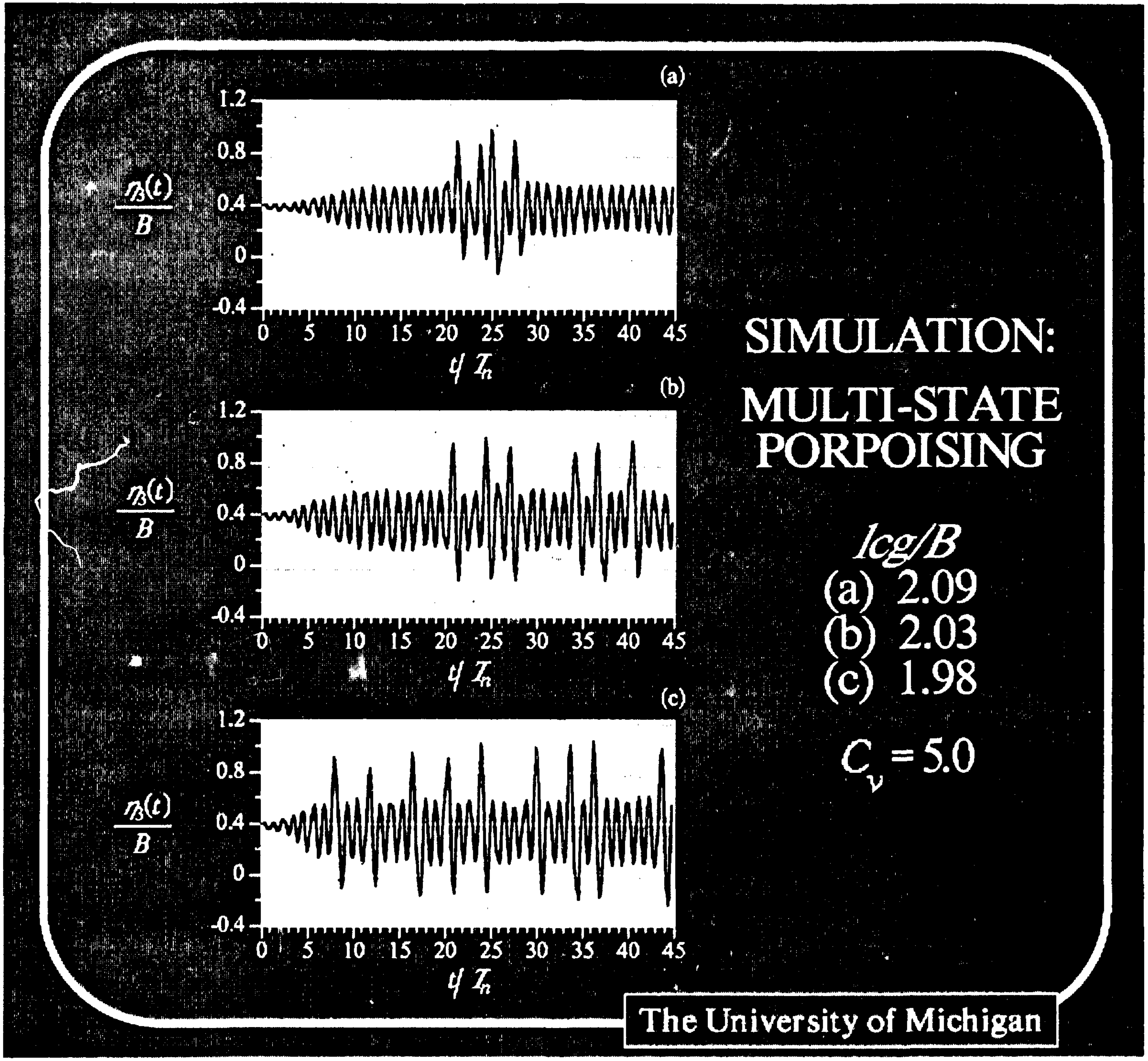


SIMULATION: TRANSITION TO PORPOISING

lcg/B

(a) 2.09
(b) 2.03
(c) 1.98

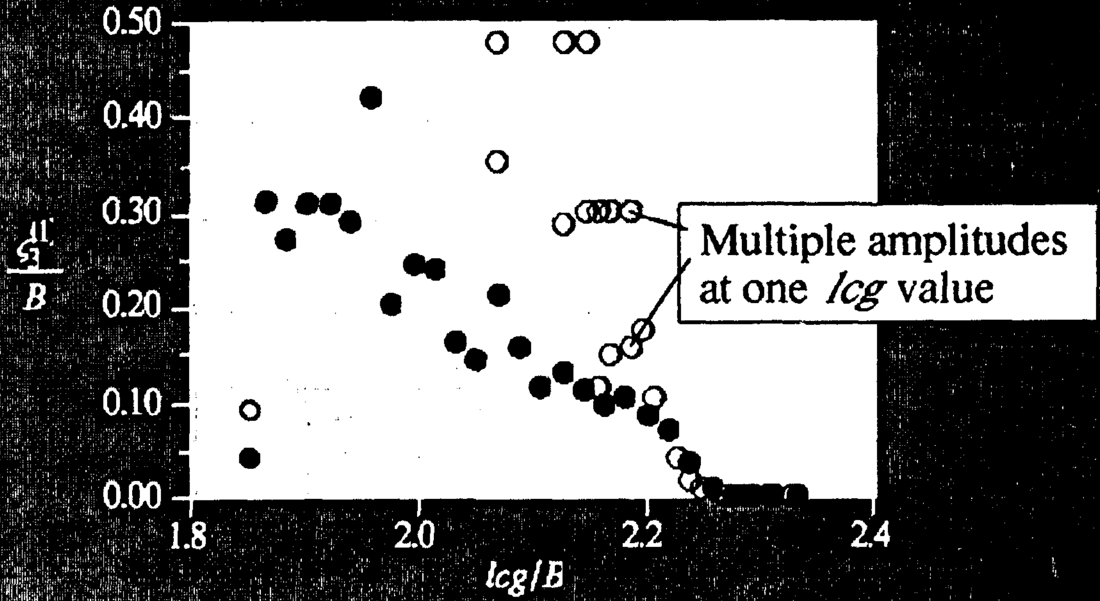
$C_v = 4.5$



SIMULATION:
MULTI-STATE PORPOISING
lcg/B
(a) 2.09
(b) 2.03
(c) 1.98
$C_v = 5.0$
(a)
(b)
(c)
$\frac{\eta_3(t)}{B}$
1.2
0.8
0.4
0
-0.4
0 5 10 15 20 25 30 35 40 45
t/T_n
The University of Michigan

SIMULATED UNFORCED MOTIONS

Heave response amplitudes (porpoising) as a function of *lcg/B* ratios. $C_v = 5.0$



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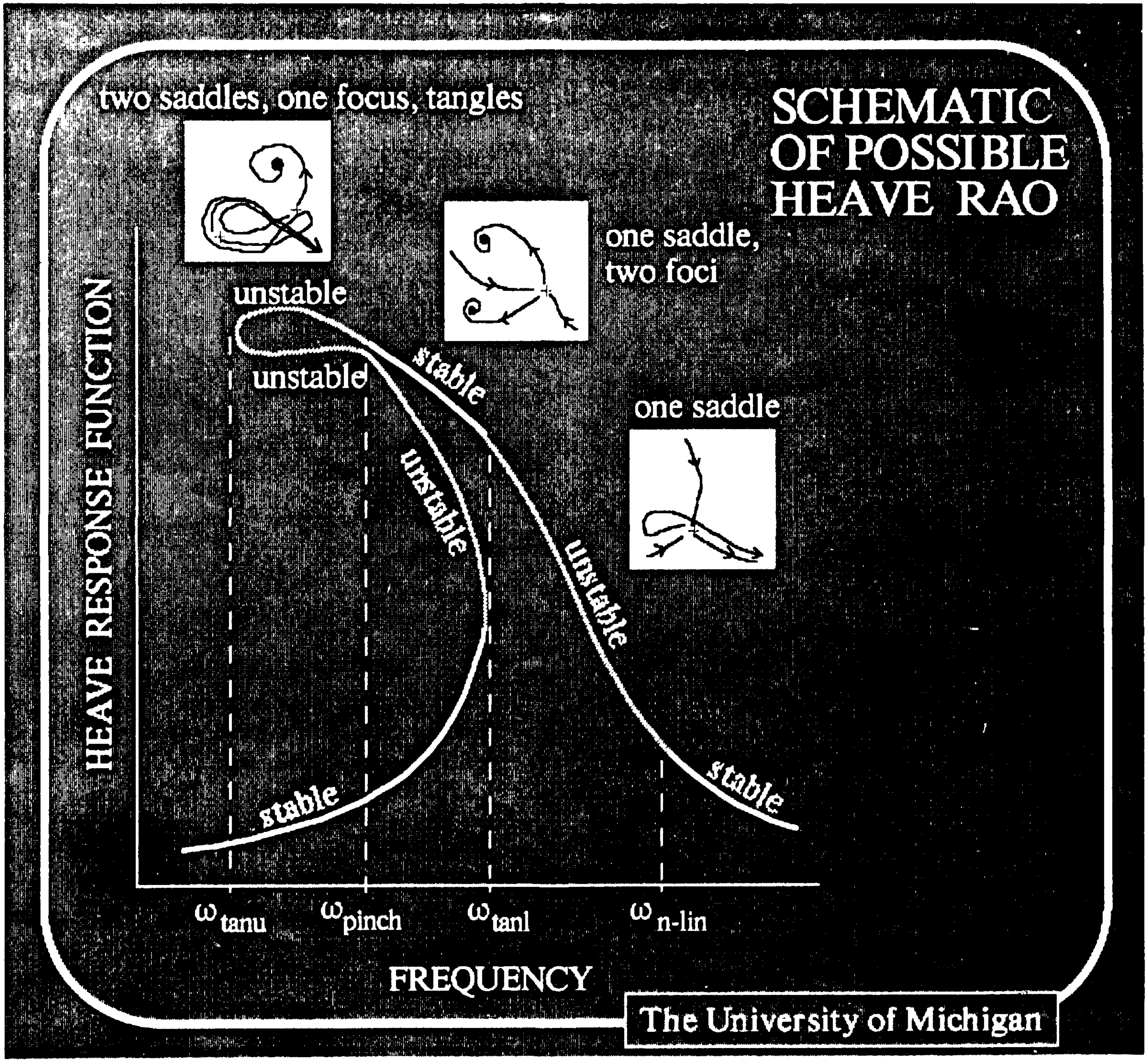
SCHEMATIC
OF POSSIBLE
HEAVE RAO
two saddles, one focus, tangles
one saddle,
two foci
one saddle
unstable
unstable
stable
unstable
unstable
stable
stable
HEAVE RESPONSE FUNCTION
ω_{tanu}
ω_{pinch}
ω_{tanl}
$\omega_{n\text{-}lin}$
FREQUENCY
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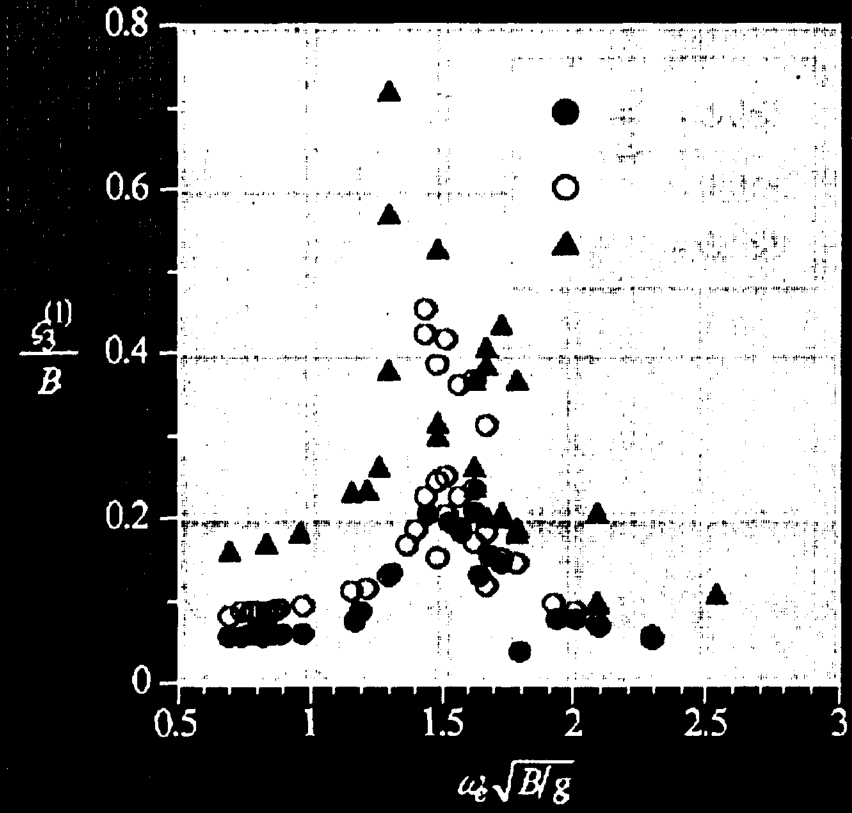
SIMULATED FORCED MOTIONS

Heave magnification as a function of frequency of encounter. $C_\nu = 4.5$



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Background on Damaged Stability

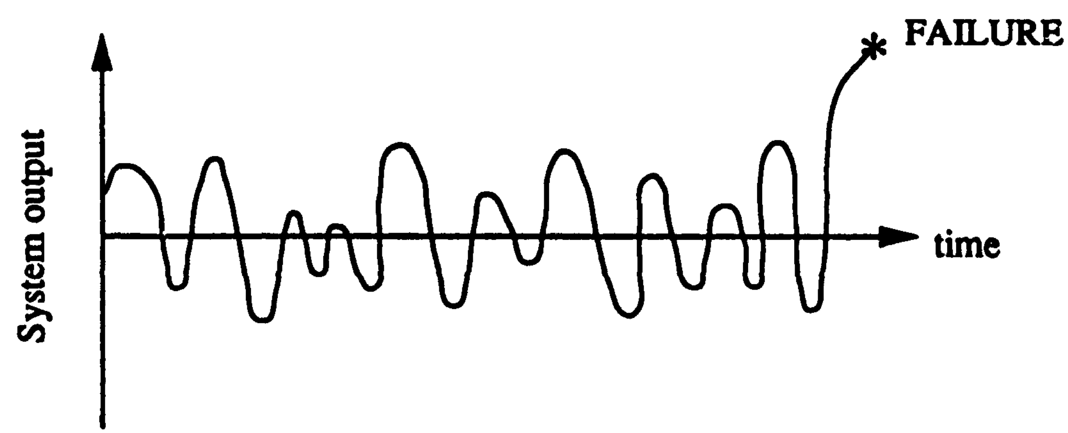
- State-of-the-art
  - USCG Regulations, USN Desgin Guidlines
  - Intact righting energy
  - Severe and gusting wind

- Nonlinear Dynamics of a Rolling Vessel in a Severe Seaway
  - Static vs dynamic analysis
  - Modeling issues

- A Nonlinear Probabilistic Approach to Extreme Vessel Motions Including Bias
  - Phase space and phase flux analysis
  - Failure associated with first excursions

A Nonlinear Probabilistic Approach to Extreme Vessel Motions

GOAL: Assessing the probability of failure for a highly nonlinear dynamical system subject to random excitation.



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A Single Degree of Freedom Model for Vessel Capsizing

Deterministic:

$$(I_{44}+A_{44})\ddot{\phi}+B_{44}\dot{\phi}+B_{44q}\dot{\phi}|\dot{\phi}|+C_{44}\phi-C_{44c}\phi^3=F_4\cos(\omega t)$$

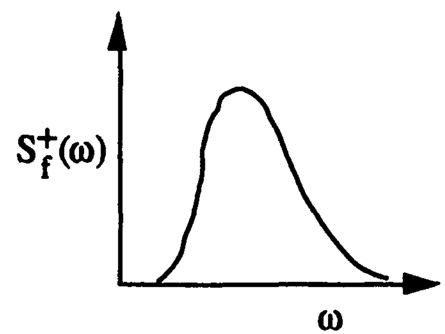
reference: Falzarano, Shaw, and Troesch, 1992.

Stochastic:

$$(I_{44}+A_{44})\ddot{\phi}+B_{44}\dot{\phi}+B_{44q}\dot{\phi}|\dot{\phi}|+C_{44}\phi-C_{44c}\phi^3=f_4(t)$$

$+C_{44_0}$ for bias

with input characteristics



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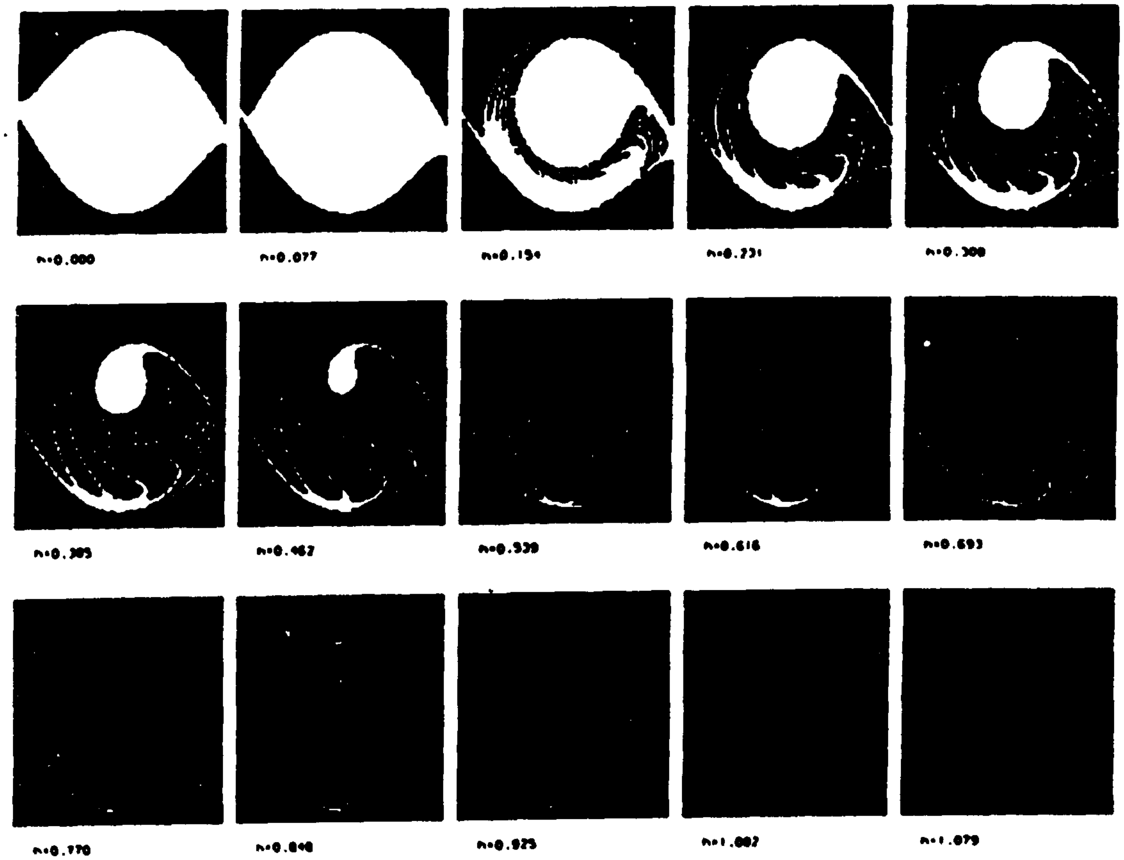
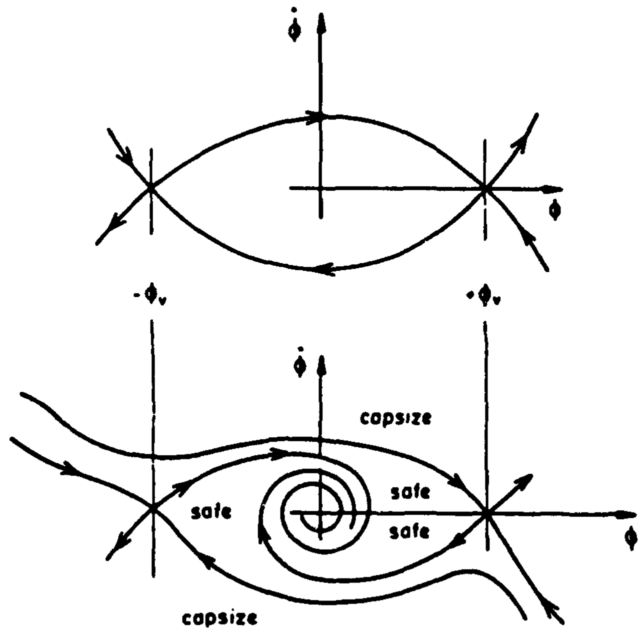
Integrity Plots for Short Time Exposure With Increasing Wave Height

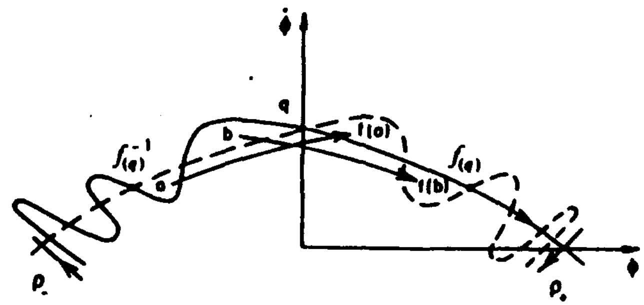


ref.: Soliman andThompson, 1991

Phase Space and Phase Space Flux



Phase planes without and with damping.

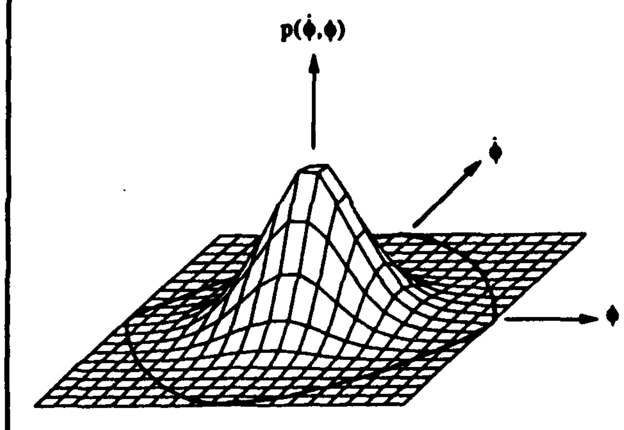


Notation for lobe dynamics.

reference: Falzarano, Shaw, and Troesch (1992)

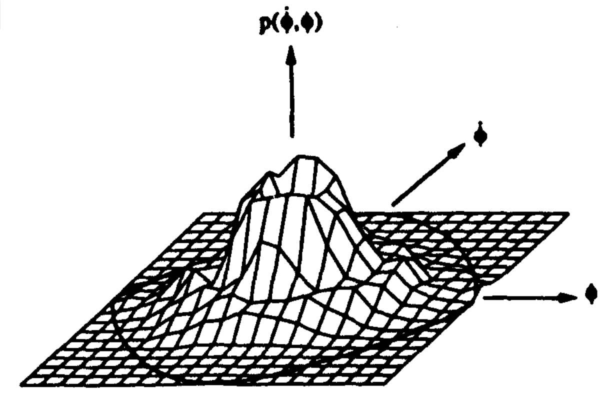
Distribution of State Variables in Phase Space Leading to Failure Based upon Simulation

Incident Sea State – ISSC Spectrum: $H_{1/3} = 30\,\text{ft}$, $T_o = 13\,\text{sec.}$



Joint PDF of displacement and velocity. Based upon 99 realizations with maximum of 1 hour exposure. Probability of capsize - 50%

Joint PDF of displacement and velocity. Based upon a single realization with capsize occurring at 12.3 minutes.



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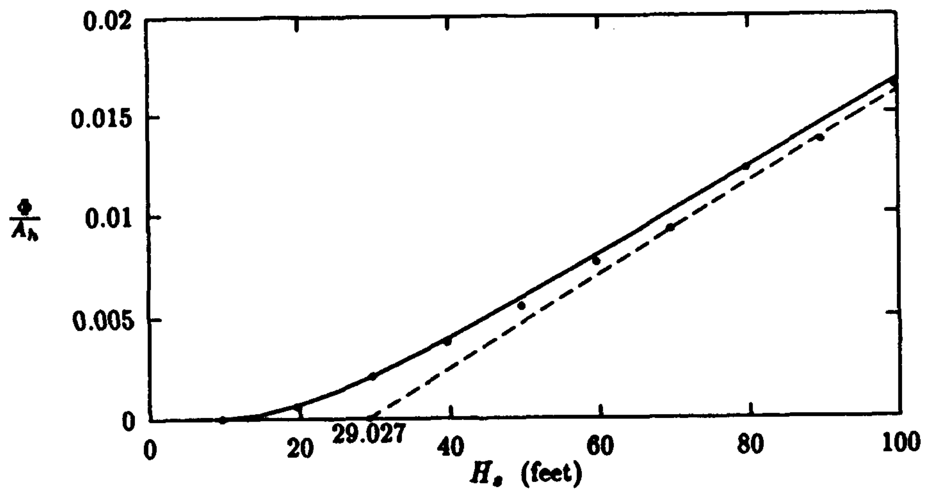
Prediction of Failure Based Upon Phase Flux Concepts



Variation of phase flux with respect to significant wave height for a 9 sec. ISSC spectrum. ---- asymptote

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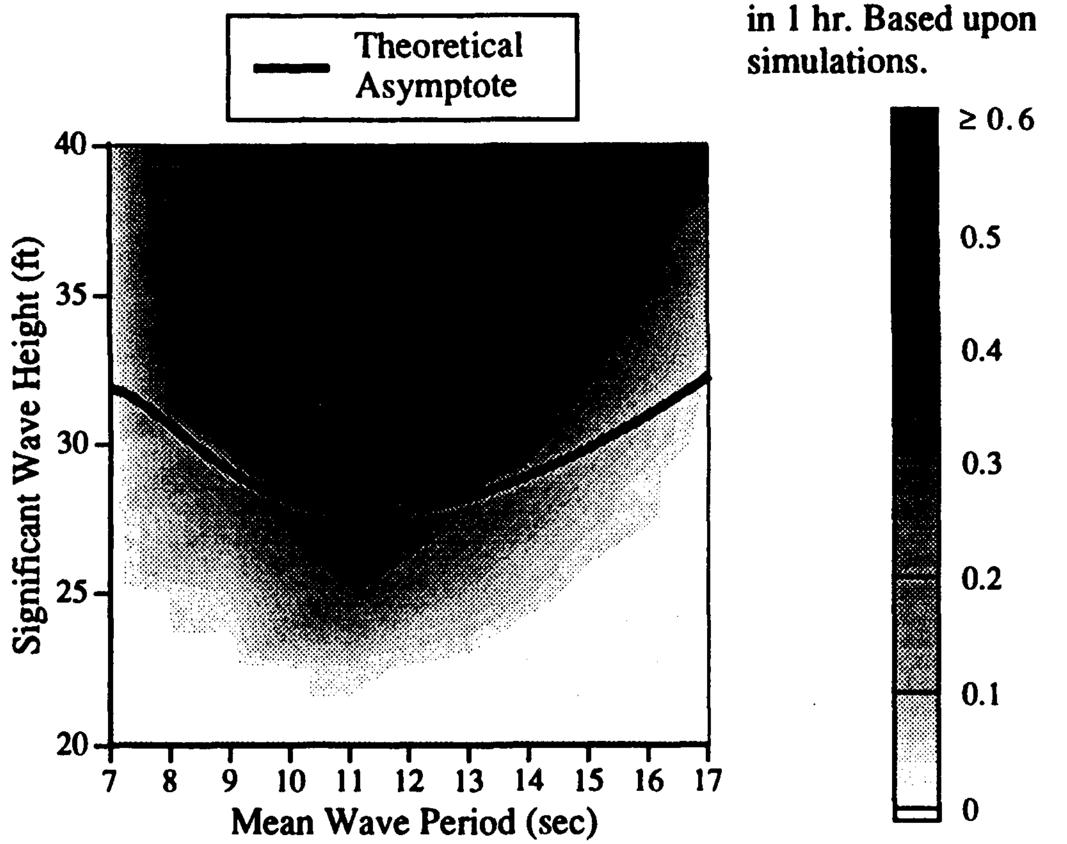
Comparison of Failure Predictions Methods Using Phase Flux Theory and Simulation



references: Hsieh, Shaw, and Troesch (1993)
and Hsieh, Troesch, and Shaw (1993)

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Summary and Conclusions

- A developing, rational technology is available for evaluating a vessel's extreme dynamics.
- The use of modern methods of dynamical systems analysis provides physical insight and increases simulation efficiency.
- Additional research needs to be done to incorporate multiple degrees of freedom, large bias and random excitation.